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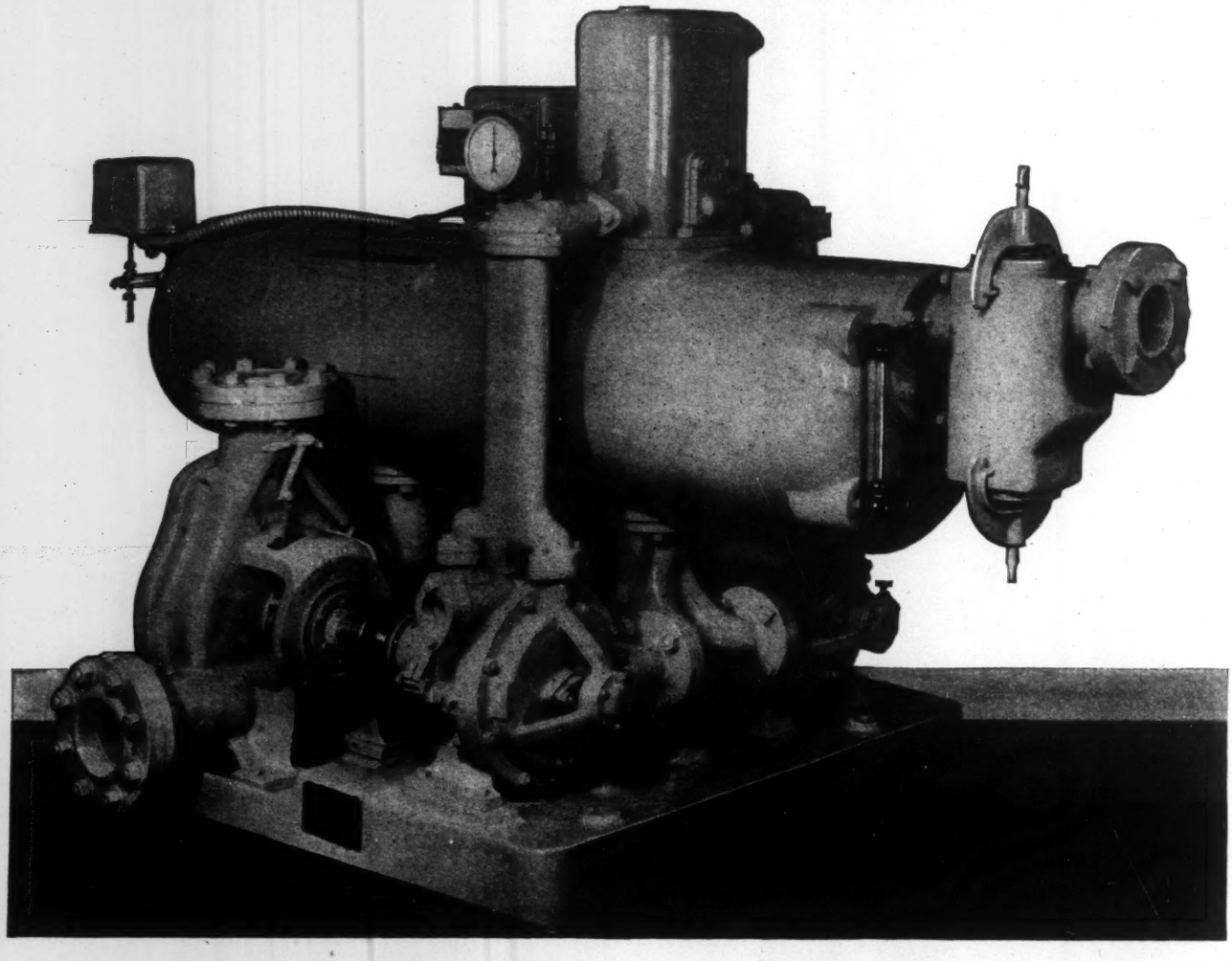

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Heating · Piping and Air Conditioning

Vol. 10, No. 8 • *August, 1938*

Dripping High Pressure High Temperature Steam Piping

By Arthur McCutchan*

As higher pressures and temperatures are used, the problem of dripping steam piping receives increased attention . . . While it has not as yet been solved with complete satisfaction, the impulse type of trap for continuous service and single orifices for drip lines operated for short periods would seem to be indicated by the investigations of The Detroit Edison Co., according to Mr. McCutchan. Strainers should precede both the orifices and the traps to prevent clogging, and open blows should be used freely during the initial operation of a piping system

THE problem of dripping high pressure high temperature steam piping has received increased attention as higher pressures and temperatures are used. While the cost of drip piping may represent only 4 to 5 per cent of the total piping cost in a modern central station, it often causes as much as 95 per cent of the piping troubles.

Ideally, a given piping system should have only one low point so the number of drips and open blows in a plant would be a minimum. However, the best piping arrangement from the standpoint of securing adequate flexibility with the least expenditure for pipe may require vertical crossovers and headers having a number of points from which suitable provision for removing condensation must be made.

In the case of saturated steam lines, condensation must be removed continuously and, where velocities are high, lines should be pitched downward in the direction of steam flow. Superheated steam lines in general require dripping only during warming up the line and when the steam flow is greatly reduced. Since superheated steam velocities are low under these conditions when condensate is present, superheated steam lines may be pitched counter to the direction of steam flow if any advantage in layout is thus obtained.

There is a tendency to provide larger drip connections, open blows, and the like on high pressure superheated steam lines than are actually needed. Of course, the practice followed in bringing a steam line up to temperature differs widely in different plants. An allowance of three to four hours is considered reasonable to bring a turbine lead up to a temperature satisfactory for starting a turbine. In warming up a turbine lead, for example, drip connections and open blows near the turbine throttle valve are opened and steam is admitted to the line through a ¾ in. bypass valve around the turbine stop valve which is located at the high pressure end of the turbine lead.

*Engineer, Engineering Division, The Detroit Edison Co.

An isometric diagram of the hot drip system installed in the Conners Creek plant of The Detroit Edison Co. is shown in Fig. 1. While on first inspection this diagram appears complicated by the number of drip lines and the valving at the traps or drip orifices, a simplified drawing eliminating drip lines from auxiliary headers, valve seat drains and the like fails to give a complete picture of the problem.

It was formerly considered necessary to provide a drip pocket close to the turbine throttle valve with a relatively large connection to the turbine lead where the steam piping approached the turbine from above. The piping arrangement shown in Fig. 1 automatically provided a large drip pocket for the 30,000 kw units. Experience has shown that such a drip pocket can be dispensed with and that a ¾ to 1 in. connection is entirely adequate. Turbine leads to the 60,000 kw units do not have any drip pockets. The method of connecting drip piping and drains to main steam pipe lines is illustrated in Fig. 2.

The practice followed by The Detroit Edison Co. is to provide a short nipple of heavy walled, ductile, low carbon pipe, Schedule 160, for 650 and 865 lb per sq in. steam lines. The nipple usually is made one pipe size larger than the balance of the drip line to care for this greater wall thickness. For example a ½ in. source valve is counterbored to receive a ¾ in. nipple at the inlet end and ½ in. pipe at the outlet end. These source nipples provide additional mechanical strength both by reason of their greater diameter and heavier wall.

Some idea of the extent to which the use of socket welded construction has simplified and improved the appearance of drip piping can be gained from Fig. 3.

While in general the standards used in drip piping correspond with those used in the main steam lines, the intermittent nature of the flow and the lower temperature resulting therefrom permit the use of composition asbestos gaskets and carbon steel valves, fitttings and flanges where the temperature conditions

in the superheated steam lines require metal gaskets and alloy steel valves and fittings. The following selections of high pressure drip piping for the 865 lb per sq in., 910 F piping system at Delray illustrate this practice.

Pipe—Carbon-molybdenum seamless pipe, Schedule 160, A.S.A. B36.10.

Flanged Fittings, Valves, and Companion Flanges—Forged carbon steel, or cast carbon steel (Carbon-molybdenum or other alloy steel may be substituted if its weldability is acceptable). Inlet side of orifice or trap; American Standard 1500 lb S.S.P. Outlet side of orifice or trap; American Standard 600 lb S.S.P.

Gaskets—Composition asbestos gaskets, $\frac{1}{32}$ in. thick.

Bolt Studs—Chrome-tungsten-molybdenum. A.S.T.M. Specification A193, Grade B13, threaded ends only (full threaded bolts are permitted in valve bonnet joints).

Nuts—Carbon steel, A.S.T.M. Specification A194, Class 2H.

Pipe Joints—Socket welded, or flanged with $\frac{1}{4}$ in. serrated raised faces.

It will be noted from the above selections that bolt studs threaded at the ends only are used in line joints and that full threaded bolts are permitted in valve bonnet joints. While bolt studs with full size unthreaded shanks are preferable for service conditions where creep is a factor, the differences are not sufficiently large to warrant departure from the almost universal practice of using full threaded bolt studs followed by valve manufacturers. The lesser length of bolt subjected to high stress and consequent higher rate of creep is, of course, the reason for preferring bolt studs threaded at the ends only. Under conditions producing fatigue failure, as in connecting-rod bolts of reciprocating engines, the use of reduced shank or full threaded bolts is supported both by test and service experience, but in usual piping applications the greater resilience and lesser stress concentration obtained with such bolts is of little practical significance.

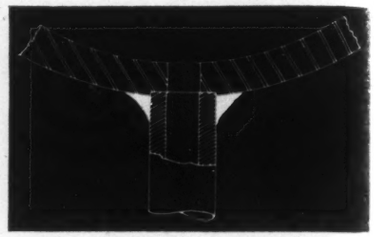


Fig. 2—Method of connecting drip lines to steam pipe

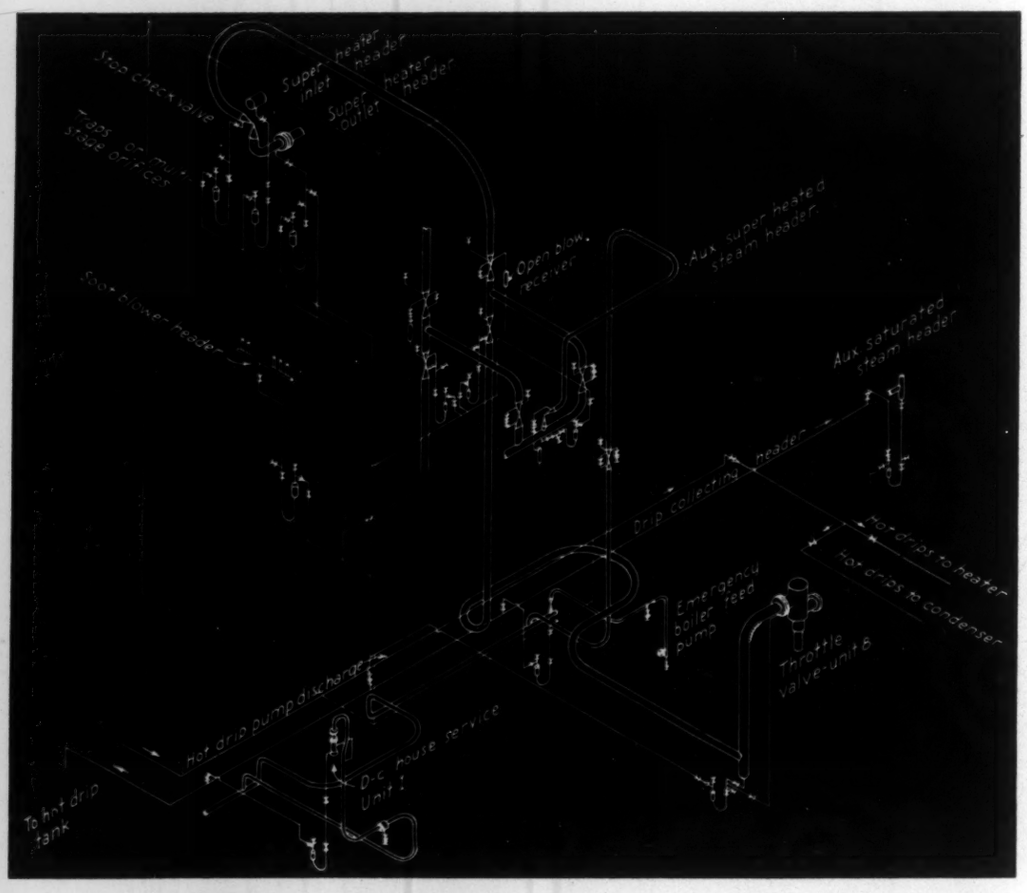
The use of composition asbestos gaskets under service conditions where temperatures in the neighborhood of 800 F may be encountered might seem questionable, but this selection is supported by some three years satisfactory experience with composition asbestos gaskets in the flanged connections to the 30,000 kw, 825 F turbines at Conners Creek.

Since we had found conventional traps unsatisfactory as a means of controlling removal of condensate from high pressure, high temperature steam lines, efforts were directed toward developing some form of labyrinth such as a number of orifices in series which would be less susceptible to clogging than a single orifice of smaller diameter. For, despite all precautions, there is always a certain amount of extraneous matter to be removed during initial operation of a plant. Experience with single orifices in 400 lb, 700 F drip lines at Delray had been reasonably satisfactory after mill scale and other dirt in the lines had worked out of the system.

The multi-stage orifice design shown in Figs. 3 and 4 made it possible roughly to double the diameter of the orifice over that for a single orifice for the same discharge capacities. As may be noted from Fig. 4, the multi-orifice design embodies a number of orifice plates spaced in a welded assembly. By selection of different numbers of plates and sizes of orifices a variety of capacities was obtained. Steam and hot water discharge capacities were determined by flow tests.

While these multi-stage orifice assemblies were found to be much less troublesome than the traps which they replaced, the amount of steam discharged increased rather rapidly with use since the low carbon machinery steel plates originally used were found to erode badly due to the action of the wet steam. Orifice plates of several

Fig. 1—Isometric of drip piping at Conners Creek plant



materials have been installed with varying degrees of success. High carbon steel plate has been found about as good as any of the materials so far tried.

Tests made with superheated steam at 600 lb per sq in. and 700 F and with the boiler feed water at 350 F showed practically the same ratio of steam to water discharge capacity for the single and multiple orifice designs. The ratio of the amount of superheated steam to the amount of boiler feed water discharged was found for the particular pressure conditions of test and orifice design as approximately one to 14. If a satisfactory strainer is developed to prevent clogging of the smaller orifices, these tests would seem to indicate that as good results might be expected with the smaller single orifice design as with the multiple orifice type. In either design the orifice should be made of material resistant to erosion by wet steam, or enlargement of the orifice with attendant increase in steam loss will result. Experiments with stainless steel and stellited orifices are under way.

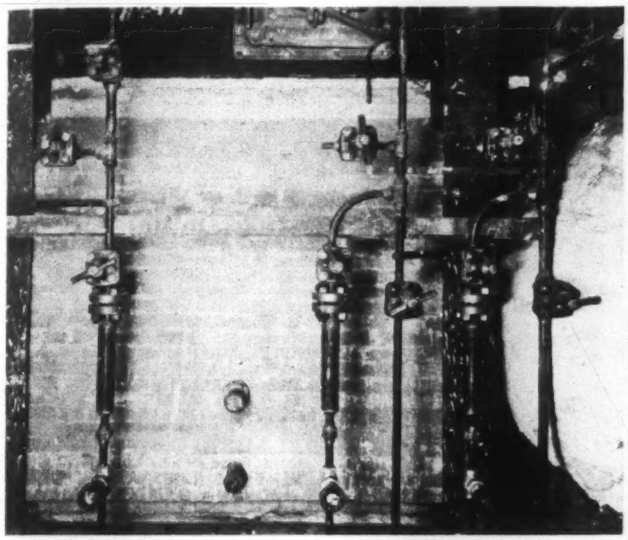


Fig. 3—Drip piping—Conners Creek plant—multi-stage orifices

While orifices, despite their limitations, had given good service, something better than either a simple or multiple orifice was desired for continuous drip conditions as the ratio of amount of steam discharged to the amount of condensate that can be passed is rather high. Where the drip lines are open only in bringing the line up to temperature, orifices may be used without any particular loss of heat.

A type of trap developed for high pressure service which is described as an impulse trap is illustrated in Fig. 5. In this type, flashing of the hot condensate tends to force a small piston into the discharge opening when the temperature of the condensate approaches within about 30 deg of the saturation temperature. As soon as condensate collected in the drain system cools sufficiently below the flash temperature, the trap opens and discharges accumulated water until the temperature of the condensate once more approaches the saturation temperature and flashing closes the trap, again repeating the cycle. A small orifice permits a continuous discharge of steam or flashed vapor when the trap is closed. The

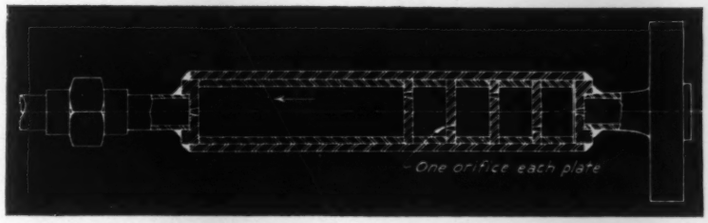


Fig. 4—Details of multi-stage orifice

operation of impulse traps was described in detail in *Mechanical Engineering*[1], August, 1936.

Tests of a ½ in. and a ¾ in. impulse trap with superheated steam and with boiler feed water showed that when properly adjusted the ratio of superheated steam to hot water that could be discharged was approximately one to 85. The conditions of test were the same as used in testing the single and multiple orifice designs. Since the operation of the trap was found to be quite critically related to the proper setting of the control chamber, it may be desirable to have the steam and hot water handling capacities of these traps checked by flow tests either by the manufacturer or in the user's plant to insure proper adjustment of the control chamber.

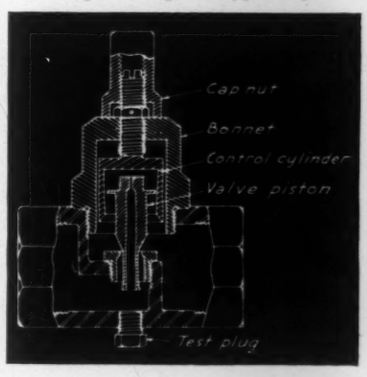
A recent development of this type of trap which incorporates a flanged strainer and flanged bonnet is under test. One month's service on a 630 lb, 825 F auxiliary superheated steam header, where because of small flow considerable condensation must be removed, indicates that good results may be expected from this trap. The ratio of amount of superheated steam to hot water discharged, in this case, was found to be approximately one to 200.

The selection of body and seating materials for such service is of course a matter of trial and correction. Trap materials satisfactory for drips from a 650 lb, 850 F line (for example) may not stand up in service at 865 lb per sq in. and 910 F. Only prolonged service will finally demonstrate the adequacy of the materials selected. Carbon-molybdenum bodies, stainless steel control cylinders and strainers, and stellited valves and seats are used in the trap under test.

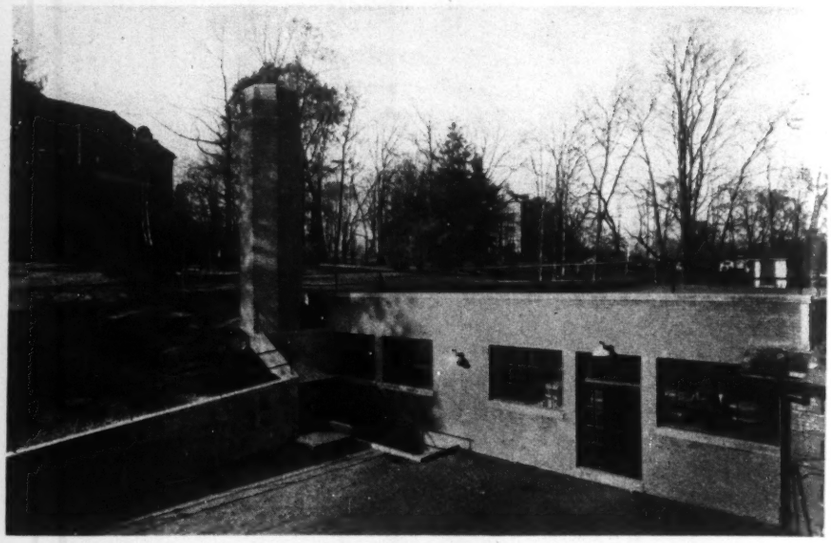
The problem of dripping high pressure, high temperature steam piping has not been solved with complete satisfaction but the impulse type of trap for continuous service and single orifices for drip lines operated for short periods would seem to be indicated by the investigations made so far. Strainers should precede both the single orifices and the impulse traps to prevent clogging the small passages. Open blows should be used freely during the initial operation of a given piping system.

The author wishes to acknowledge the assistance given in the preparation of this article by Messrs. J. A. Blecki, W. W. Brown, A. C. Pasini, and C. R. Schmansky of The Detroit Edison Co.

Fig. 5—Impulse type trap



[1] "Fluid Flow Through Two Orifices in Series," by Milton C. Stuart and D. Robert Yarnall.



The central heating plant and the high velocity stack

Performance of Mount Holyoke's Heating Plant

By D. W. McLenegan*

When Mount Holyoke College installed a multiple-unit central heating plant four years ago comprising 120 oil fired boilers under complete automatic control a great deal of curiosity and considerable criticism was aroused. How the scheme has worked out in practice is described here, and the reasons for adopting it are summarized. . . . While the author does not intend to put forward the multiple boiler idea as the best solution to large scale heating problems in general, he feels that Mount Holyoke's choice for the particular situation that they faced has been amply justified

FOUR years ago a central heating plant of novel design was installed at Mount Holyoke College, South Hadley, Mass., consisting of 120 oil burner-boiler units, all situated in one boiler room, and furnishing steam through existing underground mains to 29 of the principal buildings. In addition, 51 units of the same design were installed either singly or in small groups to heat outlying buildings not connected to the steam distributing mains.[1]

Since the use of so many boilers in a central heating plant is unusual, many questions have been raised regarding the performance of the plant and some criticism has been encountered of the choice of so many small units. It is now possible to summarize the operating results of this installation and to set down specifically the savings which have resulted and the low record of maintenance and repair this installation has established.

The design *is* unusual; no general statement can be made regarding its applicability to other heating loads of the same magnitude. Therefore, it is worthwhile to repeat briefly the considerations which led to the selection of this plant for this particular situation, and which have proved advantageous during the operating period since installation.

1. *Old Plant Obsolete*—The earlier hand-fired heating plant, after 30 odd years of service, was to be replaced by a more modern plant with automatic operation.

2. *Chimney Appearance Objectionable*—The existing plant was unsightly and, from the architectural planning standpoint, its elimination was highly desirable.

3. *Difficulty of Relocating Plant*—This led to consideration of a new plant, to be situated about one-half mile from the old location. However, a change of plant location would have involved difficulty in relocating the underground steam mains. The campus is on hilly ground, underlaid with rock, hence the addition and rerouting of steam mains would have been very costly.

4. *No Space for Individual Plants*—In most of the principal buildings, basement spaces were already in use as class rooms, laboratories, etc., leaving no space for individual heating plants. Further, many of the buildings did not have chimneys, which could have been added only with considerable sacrifice of symmetry and architectural effect.

5. *Plan for Using Original Location*—Back of the old plant, and partly embedded in a hillside, was a large coal bunker. By using this space for a new boiler plant, and with equipment requiring only small headroom, it was possible to build a new and inconspicuous plant without changing the location.

Through the ingenious solution of the problem which was adopted, the appearance of the campus was improved, while at the same time the expense of new steam mains was avoided. Today, grass is growing in the sod laid over the roof of the new plant, which is invisible from nearly all parts of the campus. A 20 ft stack of simple design extending above the plant carries away the flue gases which leave the induced draft fan at high velocity; but neither noise, smoke, nor fumes betray the operation of the boiler plant below.

At first, it was planned that the coal bunker structure, after being cleaned and repaired, should house the new equipment. But before construction was started, the decision to build a new library on the campus, to be heated from the central plant, necessitated an enlargement. Consequently, the coal bunker was torn down,

*Air Conditioning Dept., General Electric Co.

[1] A description of the installation, by B. R. Sheldon and F. A. Westbrook, was published in the November, 1934, HEATING, PIPING AND AIR CONDITIONING,

and a slightly larger building erected in its place to house the heating plant. The new building was heavily constructed so that the space above might be sodded and used as a platform for college activities.

How System Is Controlled

Modern automatic heating plants were also installed in the buildings not heated from the central plant. Many of these buildings are residences requiring only single boilers. The largest separate installation, involving 11 boilers, was made in Mandelle Hall dormitory. Here the small boilers were placed in a space broken up by supporting columns, and the former boiler room was made available for other uses. In this building, four separate heating zones were established, each with thermostatic control because of the differences in exposure of the various faces of the building.

All boilers in the Mandelle plant are connected to a single steam main, and the branches are sectionalized by motor operated valves.

Upon a call for heating in any zone, the proper steam valve opens and the corresponding group of units is automatically started. To provide quick response when heat is needed, the control automatically provides for starting one additional unit if none of the other boilers happen to be under fire. For mild weather, a single unit with indirect heater and storage tank supplies hot water for the building, and the other 10 units are shut down, to eliminate completely the standby loss.

The heating of any building served by the central plant is regulated by remote control from the central plant. By this means also, the heating up of various buildings in the early morning is accomplished in steps, so that the boiler load is brought up in small increments.

Where one pipe steam radiation is installed in a large building, there is always a problem of supplying uniformly a small quantity of heat in mild weather. Since all of the principal buildings had one pipe systems, this problem was dealt with by a special modification of the control system. A heating cycle of about 10 minutes was established for all buildings and steam is admitted to each building by a master valve for a variable percentage of the 10 minute cycle up to a maximum of eight minutes. This adjustment is automatically controlled from the central plant in accordance with outdoor temperature, although the control enables the operator to meet unusual heating requirements (such as the case of a special evening meeting) in any building.

Performance of the Plant

The first winter—1934-35—was to some degree a period of trial and adjustment, although the plant delivered adequate heat at all times. Since that time, the plant has operated without failure, and with small expense for maintenance. When desirable, boiler units are occasionally withdrawn from service for inspection or adjustment without noticeably affecting the output or operation of the plant as a whole. Thus any necessary servicing can be done at the convenience of the operators, without requiring immediate or emergency attention.

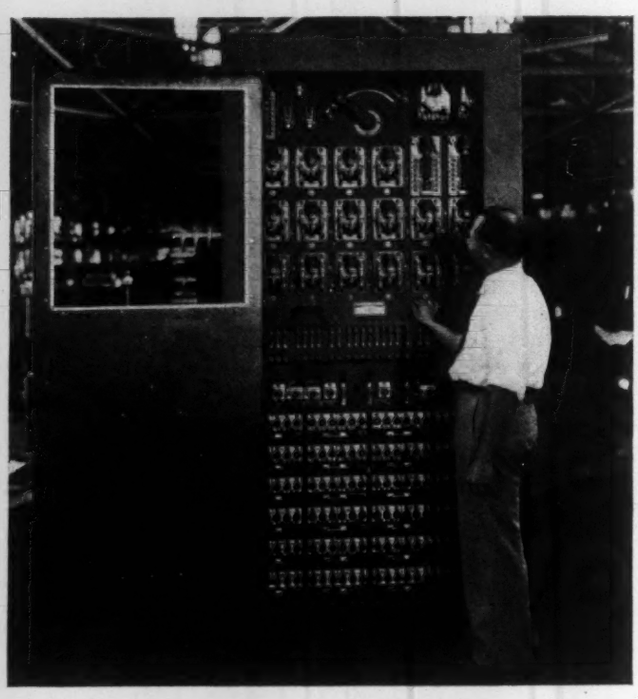
For the 1935-36 heating season, the operating records of the plant engineer have been carefully examined in cooperation with authorities of Mount Holyoke College. By comparison with the old plant, and with the former operating costs corrected for weather variations and for changes in fuel costs, an operating saving of $16,000 for the season was disclosed, taking into account the various isolated plants as well as the central plant. The saving consists largely of reduced costs of fuel and attendance. Although many of the isolated buildings are considered to be more adequately heated with the new equipment, no attempt has been made to adjust costs to include this point. The comparison does, however, include all repairs to the equipment, insurance costs, and the cost of electric energy for plant operation.

Of equal importance was the saving in first cost of the plant as compared with the cost of the alternative involving a plant at a different location. Based on studies made before the final selection, this investment saving was approximately $100,000, according to figures approved by the College.

Of particular interest to engineers are the results of the monthly summaries of boiler plant efficiency, recorded and summarized by the plant engineer, B. R. Sheldon. The record for September, 1935, shows an overall oil-to-steam efficiency of 76.5 per cent for the month's total operation. This figure is unusual in that it represents a month's operation at an average load of less than 20 per cent of plant capacity. During the other months of the 1935-36 season, the monthly summaries showed efficiencies ranging from 75 to 78 per cent, the weighted average for the season being 77 per cent—an unusually high average. The same efficiencies and low maintenance have been recorded for the 1936-37 and 1937-38 seasons.

Table 1, extracted from Mr. Sheldon's annual sum-

Some of the 120 oil fired boiler units in the plant



Master regulator for the 120 units

mary for the 1935-36 heating season, indicates further the overall performance of the heat generating and distributing system.

The annual figure of 5.18 gallons of oil per sq ft of radiation is of interest because it includes several unusual items. First, there is the fixed loss, continuous throughout the heating season, from the existing underground steam mains. Second, there is the rather long-day occupancy of college buildings, many of which are used as dormitories and study halls, in addition to the classrooms. Also included in this figure is the water heating load for the various buildings.

The figure of 6.13 gallons of oil per sq ft for the separately heated buildings is an average of figures varying over a range of more than two to one, and this variation indicates the extent to which heating costs may be affected by the dictates of the occupants even where the heating equipment and the method of thermostatic control are uniform throughout.

Kind of Fuel

In discussions of this plant, the point has been raised that the use of larger boilers with Bunker C oil would have reduced fuel costs. This is true insofar as the central plant is concerned. This plan was carefully studied in advance, and was found to have the following deficiencies:

1. Separate oil supplies would have been required for the central plant and for the isolated buildings. This in itself would have raised the oil delivery costs, since oil is delivered by motor trucks and frequent cleaning of the tank trucks would have been necessary if both No. 6 and medium grade furnace oil had to be transported.

2. In addition, the use of the large central storage tanks as an emergency oil supply for the outlying buildings—a valuable safeguard—would have been sacrificed.

Table 1—Fuel Consumption

Item	Central Plant	Separate Buildings
Load, EDR	154,545	38,086
Degree days	6,395	6,395
Average temperature of feed water, F	155	....
Efficiency, per cent	77.0	....
Days in season	302	302
Gallons oil per sq ft	5.18	6.13
Gallons oil per sq ft per D.D.	0.00081	0.00095

3. The centrally heated buildings represent less than 65 per cent of the radiation load. Hence, any saving through the use of heavier oil would have been applicable only to this part of the total fuel supply. A further offsetting factor would have been the increased rate for the furnace oil, if purchased for only 35 per cent of the load, in the buildings heated separately.

4. The burning of heavy, low grade oil requires large boiler units. Such units could not have been housed in the plant as it was built, consistent with the planning of that part of the campus.

5. Even if means had been found to use a few large boiler units in the present location, it is doubtful whether Bunker C oil could be burned without giving rise to some complaint of smoke and fumes from nearby buildings. With medium grade oil, the present plant has justified all expectations in that no complaint of smoke or of oil fumes has resulted, even though several occupied buildings are situated very near the boiler plant.

From these considerations, it is felt that under the circumstances in this case the use of small units using an intermediate grade of oil was justified.

It is not the intention to put forward the multiple boiler idea as the best solution to large scale heating problems in general. Experience based on other studies shows that many economic factors must be weighed before recommending so large a plant, although many multiple unit plants with five to 20 boilers are in successful operation. Nevertheless, for the particular situation encountered at Mount Holyoke, the selection of the multiple unit plant has been amply justified by its operating record.

Uniform Marking of Valves and Fittings

In the article of this title published in the July H. P. & A. C. diagrams were reproduced on p. 438 showing the typical body marking and typical identification plate for a 4 in. alloy steel valve rated at 1500 lb per sq in. steam service pressure at a temperature of 900 F. In the caption it was stated that all markings except items *b* (bill of material number) and *c* (serial number of valve) on the identification plate are mandatory.

Actually items *a* (size of valve) *b*, *c*, and *h* (atmospheric temperature liquid pressure rating) shown on the identification plate are not mandatory, but are optional with the manufacturer, although the size marking (item *a*) must appear either on the valve body or the identification plate. All other markings shown on the identification plate are mandatory for a valve of this type. For a carbon steel valve, however, having a steam service rating of 1500 psi at 750 F, the temperature marking (item *k*) and the alloy steel symbol in item *d* are not required.

Department Store Cuts Conditioning Cost by Careful Planning for Economy

By Walter A. Henne*

Pogue's cooling tower features the air conditioning system of which it is a part

In the design of a large air conditioning system, it is important to co-ordinate the plan with the specific requirements of the building and its existing mechanical plant. Through careful preliminary study of its ventilating system, boiler installation, and electric generating equipment when it was decided to condition the H. & S. Pogue Co. department store, very worthwhile operating economies have been achieved. . . . Mr. Henne, the operating superintendent, analyzes the problem and reports the results

IN the design of a large air conditioning system for a department store it is important to obtain the proper preliminary information and then to carefully analyze the data. Most large stores have existing ventilating systems which can be revamped and incorporated with the air conditioning system. Some have existing steam boilers and electric generating equipment. In the preliminary survey for the H. & S. Pogue Co. department store, 4th & Race Streets, Cincinnati, Ohio, all three of these were available.

Our store is an eight story building, approximately 200 ft long by 195 ft wide, with an eight floor annex in the Carew Tower Building 105 ft long by 100 ft wide. It was desired to air condition the basement and first three floors of the building, together with the beauty parlor on the sixth floor. This represents an air conditioned area of approximately 157,600 sq ft. These floors include the downstairs store and also the ladies ready-to-wear departments on the third floor. In the ladies ready-to-wear departments air conditioning is particularly desirable because it lessens soilage of the garments due to customer handling and try-ons, and decreases the resultant mark-downs. Not only in this department but in all departments, the exhilarating freshness of the conditioned air tends to brighten the salesgirls and make them more alert and efficient. Not only are the salesgirls pleasant, but the customers are more comfortable and in a better buying frame of mind. The more comfortable our customer, the longer he or she will remain in the store and in the presence of the displayed merchandise.

The sixth floor beauty parlor has 40 booths, and was a distinct design problem in itself. In a beauty parlor intermittent operation of the hair curlers, dryers, and other electrical appliances requires that the system have considerable flexibility in air temperature control and in air distribution. For example, if a booth were comfortably cooled when a hair dryer was not in operation, then when the hair dryer was put into operation and if the additional required cooling were not supplied to the particular booth, there would be a tendency for the temperature to rise and for the air to become uncomfortably warm. This requires a design for individual booth air control, so for this space adjustable air outlets were installed for each booth within easy reach of the operator. An overhead air distribution system was installed to maintain comfortable conditions in the beauty parlor during all times when the special heat loads from the dryers, etc., are not in operation. The individual booth supplies were designed to take care of only this additional heat load.

In the design of the air conditioning system for the main store, the heat loads were first calculated to determine the air quantities and the refrigeration tonnage. The required zoning of the conditioned space in accordance with localized heat loads was then analyzed. For example, in the downstairs store on sale days, as a result of crowds in this area, the air has to be supplied cooler than to the rest of the store. Also, the sun shining on the west windows of the building requires cooler air introduced to this section to maintain comfortable conditions and an even temperature than the air introduced into the interior sections of the store. There are certain partitions which separate one section of the store from another which make it desirable to zone these sections from the rest of the store.

As a result of this analysis of the people, lights, and exposure load, it was decided to make the basement a separate zone; to divide the first floor into five zones independent of the other floors; and to divide the second and third floors into five zones each. The air for the second and third floor zones was to be supplied from individual sets of apparatus placed on the second floor.

*Operating Superintendent, The H. & S. Pogue Co.

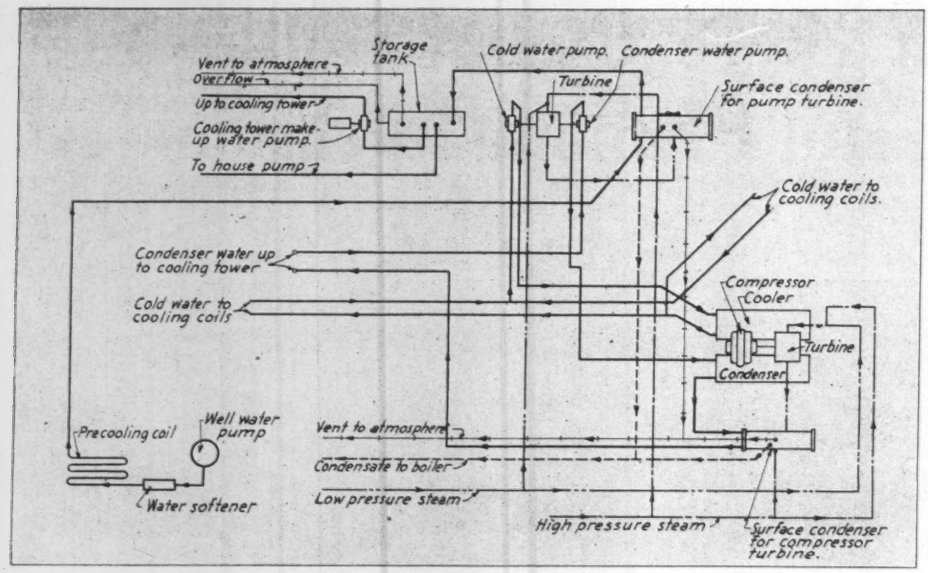


Diagram showing the steam and water flow for the air conditioning plant

To divide the store into these zones was desirable for several reasons. It meant that the equipment would be small and could be suspended from the ceiling of the store to conserve valuable floor space. It pointed toward good distribution of the conditioned air throughout the conditioned space. It meant that ducts would be small and unobtrusive. The decentralization of equipment for zones meant that the air to any one section of the building would not travel far through the ductwork.

Preliminary Survey of Existing Plant

The preliminary survey indicated the following available equipment—

1. A ventilating apparatus for the main store first floor and basement, with the supply air ducts running through trenches beneath the basement floor, rising at the columns to supply the basement, and on up at the outside walls of the basement beside the show windows to supply air from over these show windows to the first floor.

2. A ventilating apparatus located in the basement of the Carew Tower Annex, supplying air through a system of ductwork to the first, second and third floors of this annex.

(a) Each of the above two sets of apparatus took *all* outside air through heater coils, on through a spray washer and reheater coils, and supplied this to the store. For economy in air conditioning it is desirable to limit the quantity of outside air to that required for good ventilation. In general, this approximates 25 per cent of the total air supplied to the conditioned space, or about one complete change of outside air in the store every 40 minutes.

(b) The air quantities supplied by the ventilating system differed from those calculated in the design of the air conditioning system. The fans were designed for the low static pressure required for ventilating systems, whereas now it was necessary to add the additional resistance of filters, cooling coils, sprays, and static pressure in the ductwork for high velocity supply air outlets.

(c) The heating coils and the ductwork were not divided into zones in accordance with the new air conditioning requirements.

(d) The supply air outlets on the ventilating system were not designed for cool air distribution.

3. A well with a capacity of 250 gpm of 60 F water was available in the main engine room. An analysis of this water showed that it was hard water and somewhat muddy. However, it might be used for precooling in a closed system where its temperature would not be raised enough to cause precipitation of the carbonates in the water.

4. Three large boilers were available with a total capacity of 1300 boiler horsepower at 150 lb pressure. The best operating efficiency of these boilers is at 175 to 200 per cent rated load.

5. The chief purpose of the boilers was to supply steam to four electric generators with capacities of 250, 250, 300 and 800 kw. The total electric generating capacity is 1600 kw. However, it was determined that the average daily electric load for lighting, elevators, and other electrical requirements in the store approximated 1000 kw. The first generator of 250 kw capacity was started at 7:15 a.m. At 8:00 a.m. the 800 kw unit was added to the line. These generators were designed for inlet steam at 150 lb pressure and exhaust steam at 1 lb back pressure.

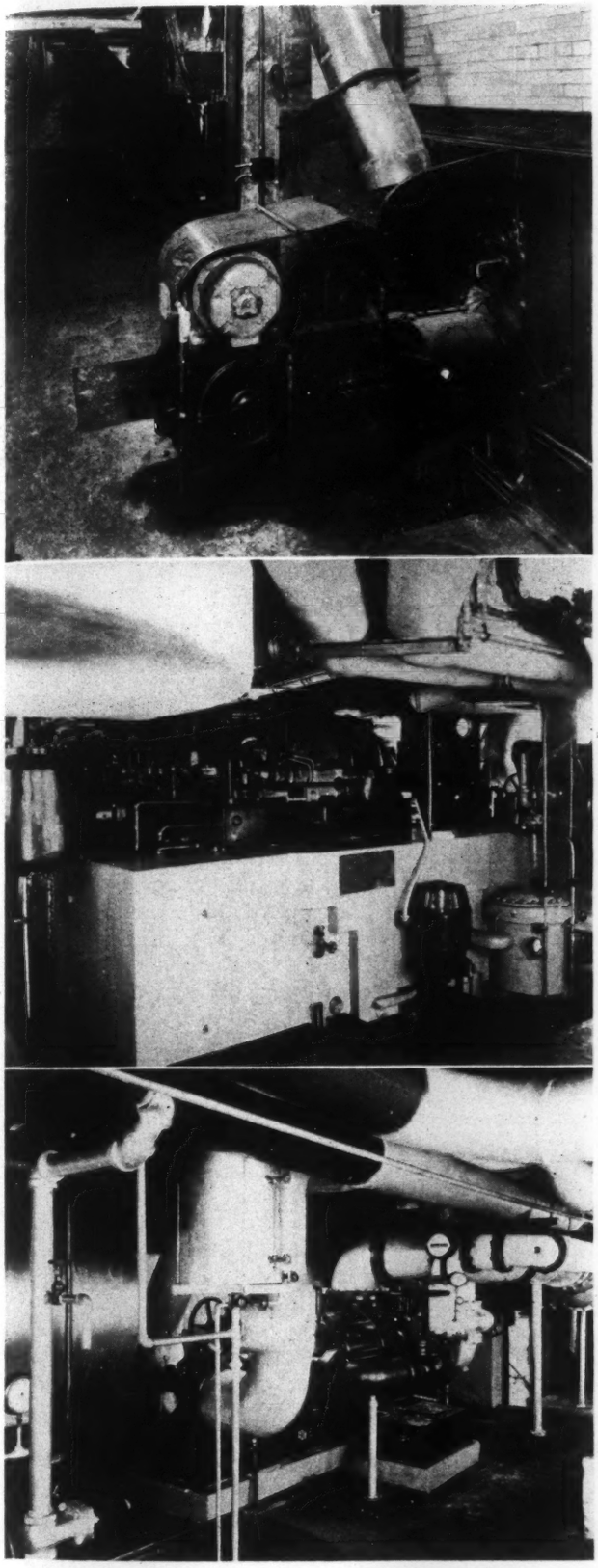
During the summer when exhaust steam was not used for hot water and heating appliances, it was wasted to the atmosphere. Less than 5 per cent of the total exhaust steam was utilized for these purposes. During the wintertime the exhaust steam besides being used to heat the water and for appliances, was used to heat the building. What steam was not required for these purposes was wasted to the atmosphere. The average steam required for the electric generators was 35 lb per kw output. With 1000 kw average load, this was 35,000 lb of 1 lb pressure exhaust steam available to be used for whatever purpose it might have in the air conditioning system.

With this basic information available, the design of the air conditioning system was started. If it were possible to use the 1 lb pressure exhaust steam to drive the centrifugal refrigeration machine and to drive the condenser water and cold water pumps, available energy in this steam which otherwise was not used would then be utilized. Also, additional electric generating equipment which would be required for these motors would not be used but would be conserved for other purposes.

Steam Turbine Drives Compressor

The refrigeration load for the basement and first three floors of the store was estimated as 435 tons. It was possible to obtain 35 tons of this by well water precooling through the main store air conditioning apparatus. For the remaining 400 tons one centrifugal refrigeration machine with a steam turbine drive was installed. The steam turbine was designed to operate at 1 lb pressure supply steam and 26 in. vacuum pressure gage. This turbine has a steam rate of 32 lb of steam per brake horsepower hour and requires at maximum load less than 14,000 lb of the available 35,000 lb per hr.

In Cincinnati, because the city water temperature rises



Top to bottom—Front view of the three 434 hp boilers showing the stoker feeds . . . The 400 ton centrifugal compressor with its low pressure steam turbine drive . . A 185 hp low pressure turbine drives the condenser water pump and the chilled water circulating pump (one at either end of the turbine axis)

to 85 F during the summertime, and because the city water department limits the amount of this water for condenser purposes on large air conditioning installations, it was desirable not only for these reasons, but from an economy standpoint, to install a cooling tower for condenser water. A 1600 gpm cooling tower was installed on the roof of the building and was designed for future additions if and when the air conditioning system was extended to the upper floors of the store. The condenser water piping between the cooling tower and the basement apparatus room was installed of sufficient size to handle the future requirements, which represented an increased cost at this time, but a future economy. The condenser water circulating pump requires 124 bhp. The chilled water circulating pump requires 57 bhp. The horsepower on the condenser water pump was estimated for its future load. Since these two pump motors represented considerable horsepower load, and since there was considerable exhaust steam still available, we decided to drive these pumps with steam turbines rather than electric motors. For economy in steam piping, maintenance, etc., one steam turbine was used to drive both of these pumps, one pump on either end of the turbine axis. This turbine is also operated at 1 lb pressure steam and 26 in. vacuum, and requires 32 lb of steam per brake horsepower, or a total additional steam requirement of 5800 lb. This left still available for future use 15,200 lb per hr of exhaust steam.

The water from the cooling tower passes first through the condenser of the refrigeration machine, then through the steam turbine condenser for the drive of the refrigeration machine, and from this point on back to the tower for evaporative cooling and recirculation.

The 250 gpm well water available is used first for precooling through the basement and first floor apparatus through extended surface precooling coils. This water passes from these coils through the steam condenser for the turbine which drives the cold water and condenser water circulating pumps. From this condenser the well water goes to a collecting tank, from which tank 90 gpm is pumped to the cooling tower for make-up water. This is the maximum quantity that is required for make-up water to the cooling tower. The remainder of the well water overflows from the tank to the sewer. Because of a muddy precipitation apparent in the well water when it went through the cooling tower and came into contact with the air, it was necessary to install some sort of chemical treatment. This problem has not been entirely solved but progress is being made.

Ducts and Outlets

After the air quantities and tonnages were determined and the analysis made to divide the air conditioned space into zones, the existing ventilation duct sizes were compared with the duct sizes required. On ducts that were too large, restrictions were used or allowances were made for less static pressure drop. On the ducts which were too small replacement ducts were installed. On the first floor, zone fans were located over the show windows in the furred space. Reheater coils were installed with each of these zone fans to give individual zone control with reheat, both for the summer and for the winter. A reheater was also placed in the basement supply duct to zone this space separately from the first floor.

Supply air ducts were run from the zone fans through the conditioned space for distribution of the air. These ducts were later furred in to blend with the architectural design of the building.

Special slot type ejector outlets were used to distribute the air in the main store. In other spaces double deflector diffuser type outlets were used for air circulation. Both these outlets have the property of mixing the cool air coming from the outlet with the room air at the ceiling of the room, to allow the air to circulate gently throughout the conditioned space. These outlets are a particular feature of this system.

With the proper distribution and balance of the supply air, it is not necessary to have an extended return air duct system. One main return air duct was used to bring the air back from each floor to the apparatus. Shoe shelves are built around this duct on the first floor to disguise it attractively.

The second and third floors of the store were divided into five zones similar to the first floor, and five complete sets of apparatus were installed, one for each zone. The equipment was situated on the second floor and each set supplies air to the respective second and third floor zones. The piping supplying the cold water for summer cooling is run from the refrigeration machine in the engine room up to the furred space over the show windows on the first floor, and from this space taps are taken up to the second floor to supply each set of apparatus. In this manner no piping was exposed to the sales area.

Pneumatic Control

A complete system of pneumatic temperature control equipment is used to operate the dampers and valves to regulate the temperatures in the store throughout the various seasons of the year. Each system is designed to automatically take in 100 per cent outdoor air when this air is at such temperature that it can be used to cool the store, and thereby economize on the refrigeration load.

A dewpoint thermostat operates the outside air maximum and the return air dampers during this intermediate season. During the winter the dewpoint thermostat operates the steam valve on the second row of preheaters to control the dewpoint in the apparatus and to regulate the relative humidity of the air. A thermostat in the outside air intake operates the first bank of preheaters to allow steam to enter this preheater when the outside air temperature drops below 35 F, and thereby prevent possible freeze-up of the system during the cold winter weather. The outside air damper in front of this preheater is designed to close when the fans are stopped and to open when they are started. The air brought in through this portion of the outside air intake is designed for good ventilation requirements. Room thermostats in the conditioned space operate the reheater valves to control the temperature in these spaces. As noted before, these are zoned so that an even temperature can be maintained throughout the store regardless of the season of the year. The control instruments are set for a comfortable differential between the outside air and the inside temperature.

A feature of the air conditioning system is cleanliness. Filters are situated in each set of apparatus so that all of the air supplied to the store is cleaned. In the main sets of apparatus the air passes through a spray where it is freshened and further cleaned. Less dirt and less soilage of merchandise is definitely noticeable.

Ducts and grilles blend with the store's architecture

Motor Starters

Magnetic starters are used on all of the fan and pump motors. For convenience in operation there is situated centrally in the basement apparatus room beside the centrifugal refrigeration machine, a master control panel with remote push buttons for motor starters. The entire system including the cooling tower can be started and stopped from this control point. A green pilot light shows on the panel when a particular motor is in operation. This panel board affords a quick check on the operation of the system and is a time-saver to the engineer in starting and stopping the equipment.

Saving by Using Exhaust Steam Turbines

The refrigeration machine and the cold water and the condenser water pump drives comprise the major horsepower load of the air conditioning system. These are driven by the available 1 lb pressure exhaust steam. The motors on the ventilating fans were already part of the store load. So the only additional horsepower required for the air conditioning system is that for the motors on the zone fans, the spray pumps for the coil and spray type dehumidifiers, and the fans on the cooling tower. This additional horsepower load is 142 hp which is an unusually low figure for an air conditioning installation of this size (40½ hp beauty parlor; 62½ hp cooling tower; 39 hp main store). With the use of a cooling tower and the well water for make-up water, the only cost for condenser water is the cost of the power to drive the cooling tower fans and make-up water pump, which motor horsepowers are included above.

With the use of steam turbine drives operating at a vacuum pressure, the exhaust steam which was formerly wasted to the atmosphere is now condensed and returned to the boiler as feed water. This represents an actual saving of water over previous operation of approximately 20,000 lb per hr.

An interesting comparison is that of the cost to operate

this system with exhaust steam turbine drives to the cost had motor drives been used. Another interesting comparison in operating cost is the difference between the use of city water for condensing purposes and the use of the cooling tower water. An analysis of operating costs on the present boiler plant indicates that steam is being produced at 23c per 1000 lb. With this cost of steam, electric current is now generated at the rate of 1c per kwhr.

In the Cincinnati area an average summer cooling season is 120 days, or approximately 1100 hr. The total additional horsepower required for the store by adding the air conditioning system, including refrigeration, fans, pumps, and cooling tower is 782 hp over and above the horsepower previously used on the ventilating system. Estimating an average load on the refrigeration machine of approximately 60 per cent of full load for the cooling season of 1100 hr, the average additional horsepower becomes 600 hp. If electric driven equipment were installed at 1c per kwhr this operating cost for the cooling season would be $6150. The operating cost for the system as it is now installed is $1400.

In addition to this saving, there should be credited to the system the amount of steam condensate returned for boiler feed water, at a cost equivalent to the city water rate, which graduated rate places this in the bracket of 10c per 1000 gal. Also to be credited to this turbine drive system should be the cost of the additional electric generating equipment required were motor drives used, plus the maintenance on this electric generating equipment.

The steam condensate credit amounts to $330 per cooling season. This reduces the above operating cost figure from $1400 to $1070 per season.

Neglecting the cost of the additional electric generating equipment (600 kw) we see that by using steam turbine drives, the saving in operating costs on this system is $5080. If this sum is credited to the operating cost of the electric generating equipment, the actual cost for this generation is reduced from 1c to ¾c per kwhr.

Saving with Cooling Tower

Now let us consider that city water had been used for condensing purposes instead of the cooling tower. The cooling tower is of 1600 gpm capacity and if city water were used approximately this same quantity of water would be required. This water not only condenses the refrigerant, but also is used to condense the steam from the turbine drives. If city water were used this water would be automatically throttled under light loads and an average of approximately 60 per cent of full load requirements could be figured. At 1100 hr per season, and 10c per 1000 gpm, the city water costs for the cooling season would have been approximately $6600.

The cost of the cooling tower condenser water is the cost to drive the cooling tower fans, and make-up water pump, or 62½ hp. At 1c per kwhr this cost for the cooling season is $600. In addition to this cost a depreciation value of the cooling tower should be charged against the yearly operation of this tower. Based on 15 years' depreciation, approximately $1200 annually should be added to the cooling tower water cost. With this charge included there is a saving of $4800 in favor of the cooling tower over the use of city water.

From the above figures it can be seen that the actual operating costs for the air conditioning system as installed (consisting of 435 tons of refrigeration) run only $1070 per cooling season, whereas were motor drives used instead of the turbine drives these costs would have run $6150 per cooling season, and were city water used instead of cooling tower water, this cost would have run $12,750 per cooling season. The comparison of these figures indicates the interesting and economic value of the use of steam turbine drive for refrigeration equipment where this power is available and its use advisable, and in the use of a cooling tower for condenser water.

With the present electric power consumption for various uses in the store, there remains available an additional 15,200 lb of exhaust steam for use to drive the future refrigeration equipment.

Operating the System

In an air conditioning installation of this size it is desirable to cool the store in the morning before it is opened for business. It is desirable to start the air conditioning system approximately one hour before the store opens. With the use of steam turbine drive for the refrigeration equipment and for the cold water and condenser water circulating pumps, this means that there must be available at this early hour before the store opens sufficient exhaust steam to drive these pieces of equipment.

Since the store lights are not used, the elevators are not in use, and other power loads are not required at this hour, exhaust steam is not usually available from the electric generating equipment, and some other means must be conjured to make this steam available. In our case it happens that at this time during this early morning brief interval, there is sufficient electric load to supply ample exhaust steam for the refrigeration machine and pump drives, otherwise the additional steam required would have been bled from the high pressure steam line into the low pressure steam supply to the turbine drives.

Advantages of Store Air Conditioning

This system was installed in the spring of 1937 and was in operation during last summer. After a year's experience we can think of no changes that could have been made originally which would have improved it, because it has produced the desired results. It has proved most satisfactory both from the standpoint of proper installation and type of equipment installed.

We are satisfied that the installation of this air conditioning system has been advisable and in every way desirable. Not only has it given our customers the convenience and comfort of shopping in a properly controlled temperature, but in a business way the advertising of this system tends to draw more people to the store who otherwise might have postponed their shopping trips until after the hot weather. It is an indication of progress in which the public is always interested. The salesgirls and the management receive many favorable comments, both verbally and written, from appreciative customers who have been spared from the heat of the summer, and who notice the freshness of the air the year around.

How Waterside's Piping Was Designed

By W. J. Angus* and C. A. Kelting†

Here's the story of the design and installation of the high pressure, high temperature steam piping at Consolidated Edison's Waterside Station. More than a description, in it the authors explain the problem and the piping requirements, then tell how they were worked out, the reasons for selecting the pipe materials used, the choice of stress figures, and why welding was adopted. . . . A second article will describe the welding procedure, stress relieving and examination of the welds, and the support of the piping

RECENT installations of 1200-1400 lb, 900 F topping turbines with their associated boilers at the Waterside Station of the Consolidated Edison Co. of New York Inc. presented some interesting problems in piping design—particularly in connection with the high pressure, high temperature main steam piping since materials and joint construction for the other lines (with the exception of the boiler feed discharge system) presented no new problems.

For handling steam at 900 F considerable thought had to be given to selection of proper materials to resist the effect of high temperatures upon steel. Steels operating at this temperature are considered within the plastic range and therefore the room temperature physical properties are no longer solely indicative of their strength and their ability to serve well over the expected long time periods.

The matter of piping joint construction was likewise of much concern as it was necessary to provide a type of joint which would be free from leaks and the usual troublesome and costly item of station joint maintenance. It requires but little imagination to visualize the situation involving great expense resulting from the necessity of periodic repairs and renewals to improper joints. Costly machine outages very often accompany line joint troubles.

Following the trend at the time the two topping installations were planned for on the unit basis. Each topping turbine operating independently with its respective pair of boilers is supplied with high pressure, high temperature steam through 12 in. pipe lines, one from each of its two boilers. In the case of the first unit, its two 12 in. lines are joined by means of a spherical mixing chamber, Figs. 1 and 2, the 20 in. outlet of which connects to the throttle valve of the machine. For the second unit, the two 12 in. lines are run one from each of the unit's pair of boilers directly to separate connections on the machine.

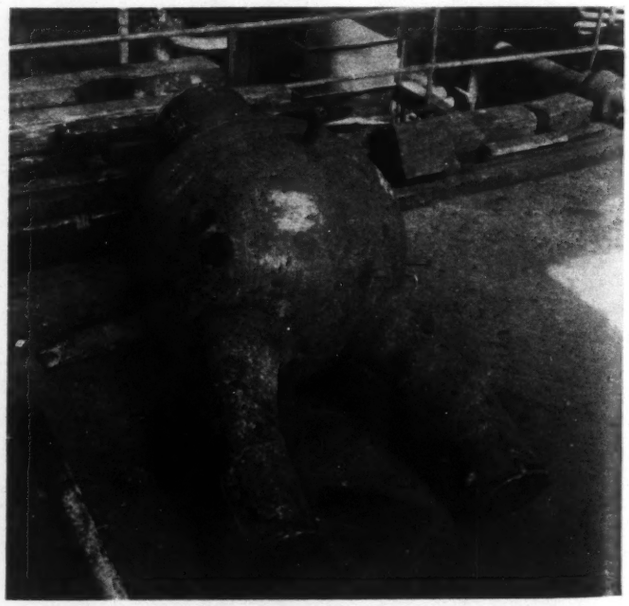
Both topping machines have their 200 lb steam exhaust connected to the station 200 lb lower pressure main steam system upon which the units are "topped" through 24, 20 and 18 in. interconnecting tie lines and headers.

Selecting the Pipe Material

An extensive investigation was made to determine what experience had been gained with steels of various kinds. The prevalent general opinion was that carbon steel is unsatisfactory for piping service temperatures in excess of 850 F. The American Tentative Standard Code for Pressure Piping, the A.S.M.E. Boiler Code and other codes and practice definitely recommend a limit of 850 F for carbon steel pipe and piping parts for steam power service. Valves and fittings of carbon steel, even at the approved adjusted pressure-temperature ratings for the 1500 lb series, are not acceptable for 1200-1400 lb, 900 F steam service. Thus, it was necessary to select an alloy steel, and carbon molybdenum was chosen as the most suitable. Research on the material indicated then as is the generally accepted consensus today that its physical properties are such as to make it suitable for use under high temperatures to 1000 F. Carbon molybdenum alloy steel behaves in many respects like carbon steel, but its qualities greatly exceed those of carbon steel at elevated temperatures.

There is a general feeling that carbon molybdenum steel is as good physically at 900 F as carbon steel is at 750 F. This leads to the assumption that valves and fittings in the dimensions of the 1500 lb carbon steel series standards are suitable for 1200-1400 lb at 900 F when made in carbon molybdenum steel of the composition given for this material in the corresponding specifications of the American Society for Testing Materials.

Fig. 1—Not a deep sea diver flat on his back, but the spherical mixing chamber or header for the steam line of the first unit



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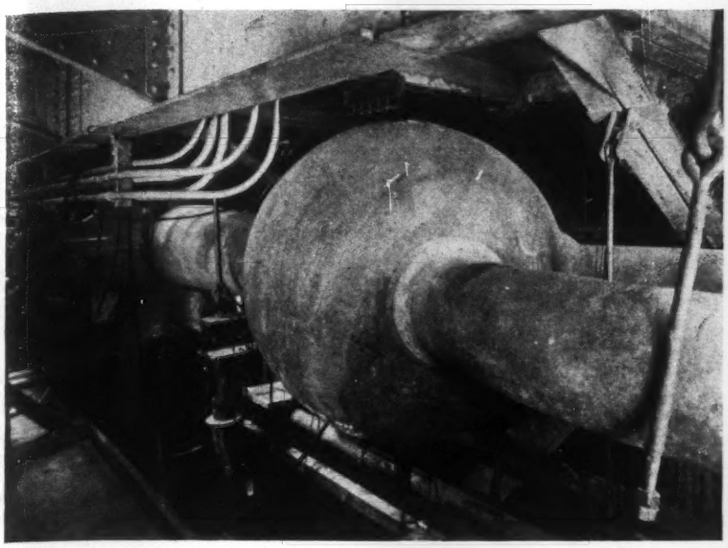

Fig. 2—The spherical header installed, the 20 in. outlet connecting to the throttle valve

Choice of Allowable Stress

In determining the pipe wall thickness as figured against bursting due to internal stress, the generally accepted modified Barlow formula was used, but a suitable value for a safe allowable stress had to be selected.

There is little doubt in the minds of metallurgists and engineers that for temperatures below 750 F, which is below the point where creep is appreciable, the working stress of equipment should bear some relation to the proportional limit of steel; but especially at temperatures above 850 F where most of the steels enter the plastic state and cease to be elastic, the maximum pressure on equipment must be based upon creep characteristics.

In this respect the behavior of the material with regard to its ability to resist creep at 900 F had to be considered. For the ½ per cent molybdenum, 0.10-0.20 per cent carbon steel pipe material, it was found that a stress of 17,250 lb per sq in. to produce a 1 per cent elongation in 10,000 hr was a conservative figure.

Collected data indicated that opinion with regard to a safe allowable stress value varied over a range from 8200 to 13,000 lb per sq in. After investigation and comparison of the properties of carbon molybdenum and carbon steel, a value of 10,500 lb per sq in. was selected for an allowable stress value for the 900 F steam piping of the first unit. This represented about 60 per cent of the creep strength of the material for design and was judged to be satisfactory; especially in view of the fact that a design pressure of 1400 lb was used although the service pressure was expected to be in the neighborhood of 1200 to 1300 lb. In solving for wall thickness, pipe of nominal wall thickness on the plus side was selected for the corresponding nominal size from the proper schedule number of A.S.A. Standard B36.10-1935 (*i.e.* if the wall thickness solved for came between two schedule thicknesses, the heavier schedule was chosen.)

Since the beginning of this work considerable interest and study of the problem has been in progress, and at the present time with revisions being made to the American Tentative Standard Code for Pressure Piping, the A.S.M.E. Boiler Code, and others, the boiler code committee has under way tentative revisions to its Table P-8 on allowable stresses for ferrous materials at various temperatures. A value of 10,000 lb per sq in. is proposed for carbon molybdenum alloy steel pipe at 900 F in this table. The reason for this lower value as compared with 10,500 lb per sq in. can be justified presumably by the fact that when the American Society for Testing Materials withdrew carbon molybdenum steel pipe grade P-1 from its specification A158 and wrote it into a separate specification A206 alone, the mills objected to a minimum tensile value of 60,000 lb per sq in. Specification A158 was the basis of purchase on our first unit main steam pipe material. The tensile value for carbon molybdenum pipe was reduced to 55,000 lb per sq in. minimum, in the new specification although it is still possible to get 60,000 lb tensile pipe. Mill reports on our first unit pipe material as a result of tests showed in the neighborhood of 62,000 lb per sq in. tensile strength.

Calculations for Wall Thickness

In our calculations for wall thickness for the 12 in. nominal pipe size, the minimum required thickness was found to come between Schedules 120 and 140, which were presumed to be standard rolled mill sizes. Therefore, from an economic point of view and on the side of safety, Schedule 140 was selected; this gave a wall thickness of 1.125 in. A recalculation based upon this wall thickness and using the same 1400 lb per sq in. design pressure resulted in an allowable stress value of 9940 lb per sq in. as figured against bursting due to internal pressure. This figure is even more in line with that proposed at present by the Boiler Code.

For the first unit, the main steam high pressure, high temperature pipe was specified and furnished where required in sizes of 8 to 12 in. inclusive in accordance with Schedule 140 while sizes under 8 in. followed Schedule 160. The supplementary requirements of the A.S.T.M. specification were called for on the order.

For the second unit, which was placed in operation June 29, 1938, main steam lines of carbon molybdenum steel in accordance with Schedule 160, having a heavier wall than was used for the 12 in. size of the first unit, were specified. This was done in compliance with an effort on the part of the prime movers committee of the Edison Electric Institute in conjunction with the steel mills to standardize on a schedule for carbon molybdenum pipe for high temperature steam service. Industry expected in doing this that the product may be rolled in quantity by the mills and placed in their stock for better deliveries. Since numerous installations were being made calling for this grade of pipe, and to prevent the mills having to roll and stock quantities in all sizes and schedules to care for intermediate services and varied opinions, it was established to set up two groups; one for services at 900 lb per sq in. and 900 F following Schedule 120 in the larger sizes and the other for services at 1400 lb per sq in. and 950 F following Schedule 160. Our service of 1200-1400 lb and 900 F, falling somewhat between

these two groups but closer to the second, necessitated our selecting the latter classification.

The spherical mixing chamber on the main steam lines for the first unit was forged in two halves from carbon molybdenum alloy steel plate and welded together. The design and fabrication was in accordance with the rules of the A.S.M.E. Code for Unfired Pressure Vessels. It has forged carbon molybdenum steel nozzles welded on for pipe and throttle connections with the proper reinforcement for structural design in accordance with the code. Fig. 1 shows this sphere as constructed and Fig. 2 shows it erected in position under the turbine.

The Boiler Feed Discharge System

The boiler feed discharge system was given special consideration. Although the pressure on the lines for this service is high—1675 to 1800 lb per sq in.—the temperature is relatively low, 376 to 400 F. Carbon steel was consequently selected. Due to the severity and importance of the service, however, a very good grade of pipe was desired and therefore Grade B seamless carbon steel pipe of A.S.T.M specification A106-36 was chosen, and the supplementary requirements of that specification for sizes 6 in. and larger were called for in order to insure better quality. The carbon content, however, was limited to a maximum of 0.35 per cent to permit welding.

Why Welding Was Decided Upon

Of great concern aside from the choice of piping materials was the matter of deciding upon the type of joint construction for high pressure, high temperature service. Although various flanged joint designs appeared feasible, the matter of bolting material presented a troublesome problem. Because piping joints are insulated, the bolting is put under temperatures very near those of the line itself. If there is any part of the piping system likely to be affected by creep to such an extent as to give trouble, it is generally felt that bolting is that part.

(Incidentally, the prime movers committee of the E.E. I., with the cooperation of a number of central stations using high temperature steam piping, has established a systematic procedure for taking periodic measurements of diameters and longitudinal lengths of pipe in the line to determine whether high temperature produces any appreciable effect upon size or shape. There is some opinion that these measurements will not reveal any changes since due to heavy walls and the service pressures involved, stresses are relatively low for producing creep).

The effect of creep manifests itself in bolting under high temperatures as the loading is more comparable to that which tends to produce creep. Here especially, it is necessary that suitable alloy steel material be used, the requisite qualities being that the material possess stability of structure and good creep resistance under high temperatures.

Bolting is customarily drawn up to give compression on the gasket or ground and lapped flange facings of about 16 times the working pressure of the line to insure a tight joint. Frequently, if a joint has been pulled up cold to produce about 30,000 lb per sq in. stress in each bolt, it has been found necessary to follow this up while hot to take care of initial expansion of the bolting material. After a long period under high service temperature, the load on the bolting may continue to decrease. This effect would be produced by creep if the bolt material, influenced by temperature, is subject to an undesirable metal structure change.

This so-called relaxation of the bolting, if the material is poor, results in a decrease in gasket pressure—which gives rise to a leaky joint. Repeated pulling up of joints, necessitating removal and replacement of insulation as well as possible repairs and renewals due to destruction of joint faces caused by the cutting action of leaks, runs the maintenance costs excessively high. This prompted the decision in favor of a full strength welded line joint to assure continued service without trouble. Oil service lines for both units were all welded to guard against hazardous conditions.

Certain joints—such as the connections of the steam non-return stop and check valves to the boilers, the boiler feed discharge system connections at the pumps and to the economizer connections of the boilers—were not welded. Flanged joints of the sarlun type with ground and hand lapped faces and without gaskets were used in these places, and it was therefore necessary to investigate bolting material for them—especially for the 900 F steam connections.

For this high temperature service, bolt studs of chrome molybdenum alloy steel fitted with 0.40-0.55 per cent carbon steel, oil quenched and tempered, hexagon nuts were decided upon. From a study of the properties of this chrome molybdenum material as compared with other commercial steels for the service, it is expected that its high creep resistance and structural stability at 900 F will eliminate the necessity for pulling up the joint after being placed in service once the joint has been pulled up cold with a stress of from 30,000 to 45,000 lb per sq in. in each bolt. Laboratory tests have indicated that the relaxation of this material produced during the initial temperature operation of the bolting does not progress far enough to reduce gasket or flange face bearing pressure to the point of joint leak, and furthermore such relaxation does not continue after the initial amount has been produced.

These bolts on the first installation were pulled up cold to about 35,000 lb per sq in. stress as calculated by micrometer caliper measurements on the stretch of the studs. After almost a year's operation, no further pulling up has been found necessary.

[A second article will describe the welding procedure, stress relieving, support of the piping, and examination of the welds.]

Data on Fans Available

The second edition of "Standard Methods Adopted for Centrifugal Fans and Blowers" is available for distribution, the National Association of Fan Manufacturers, 5-208 General Motors Bldg., Detroit, Mich. has announced. The subjects covered in this latest issue are no different from those of the first edition, but the material has been re-arranged and the comparison charts for various types of fans have been enlarged to make them more helpful to engineers, architects, and buyers of fan equipment.

Intermittent Heating of Buildings

A Study of the Fundamentals

By F. E. Giesecke*

How much additional heating plant capacity should be provided when a building is to be heated intermittently? Suggested percentages vary over a wide range, for little information on how much extra capacity is actually needed is available. . . . Dr. Giesecke believes the indefinite percentages which are now used are of sufficient magnitude to justify careful studies of intermittent heating. As a start, he analyzes the fundamentals of the problem here

IN DESIGNING the heating system for a building, it is customary to determine the maximum rate of heat loss of the building when certain required interior conditions are maintained while certain assumed exterior conditions prevail. If the building is to be heated continuously, the heating system is designed so that it can supply heat to the building at a rate equal to or slightly higher than the calculated maximum rate of heat loss of the building.

If the building is to be heated intermittently, it is customary to equip it with a heating system having a capacity for supplying heat at a rate which is higher, by a certain assumed percentage, than the calculated maximum rate of heat loss. The percentage added in this case depends upon the judgment of the designing engineer. The *Heating, Ventilating, Air Conditioning Guide* suggests that from 50 to 150 per cent of the normal heat loss of a building will be required to raise the temperature of the building to 60 F, from an initial condition of about 0 F, and that it is necessary to provide such additional capacity if a cold building is to be heated up in a reasonable length of time. Leading American textbooks suggest additions ranging from 10 to 50 per cent of the normal heat loss, depending upon the type of intermittent heating and the method of operating the heating system.

If the annual cost of heating systems installed in new buildings in the United States is estimated at $100,000,000, the percentage added to provide for the additional load due to intermittent heating may be estimated at from $5,000,000 to $25,000,000. The amount of this indefinite percentage which is added to the calculated capacities of heating systems is therefore of sufficient magnitude to justify careful studies of intermittent heating. Such studies should make possible fairly accurate determinations of the percentages to be added and thereby effect a material national economy.

It is evident that the percentage to be added should increase with the time intervening between successive heating periods. For example, in a school building in which the heating is interrupted for a period of from 2 to 2½ days, a larger percentage should be added than for an office building in which the heating is interrupted for only a few hours during any one day. For church buildings, in which the heating period may be interrupted for periods varying from 3 to 6 days, a still larger percentage should be added.

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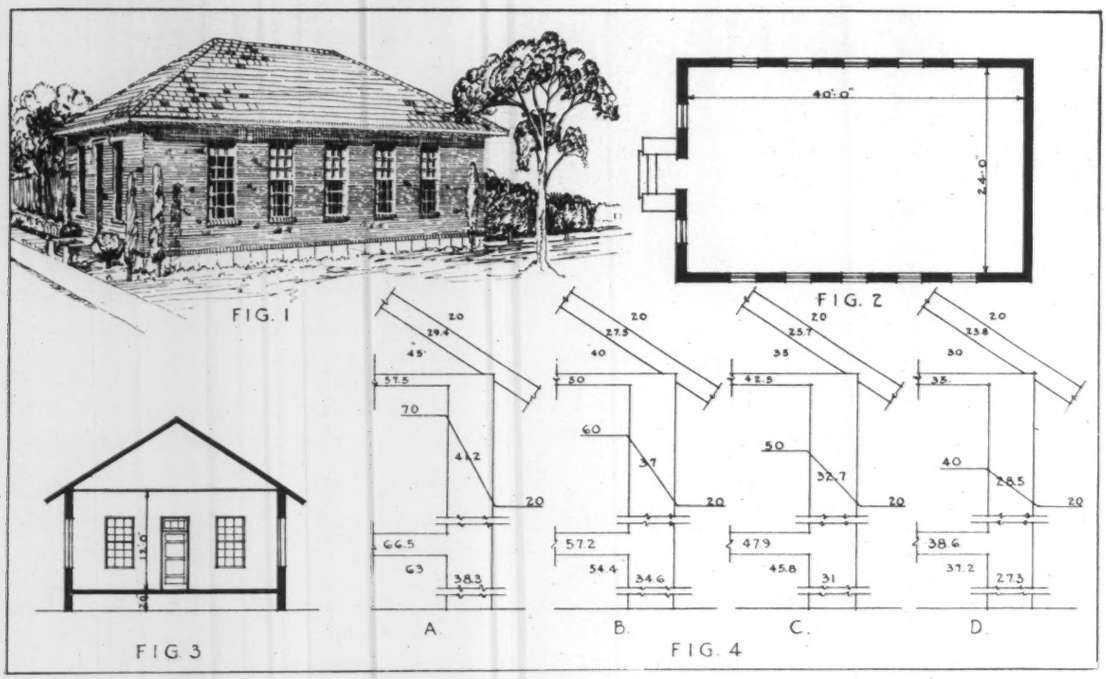
Cooling Between Heating Periods

To make a rational estimate of the required percentage, it is necessary to determine the cooling which takes place in a building between heating periods and the rate at which the building can be heated after it has been allowed to cool. Both facts can be determined only approximately because they depend upon varying exterior conditions—temperature, wind, and sunshine—and upon varying types of building construction. Refined calculations are therefore not possible and not justified.

To study the cooling of a building, consider the one room structure shown in Figs. 1, 2, and 3, and assume that it was heated until a steady state was attained when the indoor temperature was 70 F and the outdoor temperature 20 F. The building is assumed to have 12 in. brick walls with ½ in. plaster, a concrete floor 6 in. thick, a concrete ceiling 3 in. thick, and an uninsulated wooden roof covered with shingles. Assume that the heating is suddenly discontinued and the building allowed to cool while the average outdoor temperature remains constant at 20 F. Study the flow of heat from the building to the exterior and calculate the temperatures of the several parts of the building and record them on sketches as shown in Fig. 4 (*A*, *B*, *C*, and *D*) when the interior air temperature is, respectively, 70, 60, 50, and 40 F. For example, when the interior air temperature has fallen to 50 F, the average temperatures will be as follows: Space below floor, 45.8 F. Attic space, 35 F. Brick wall above floor, 32.7 F. Brick wall below floor, 31 F. Floor, 47.9 F. Ceiling, 42.5 F. Roof, 25.7 F. Calculate the weights of the several parts of the building and the quantities of heat above the assumed outdoor temperature stored in them, and record these quantities as shown in Table 1.

It appears from Table 1 that the quantity of heat lost by the building is 381,800 Btu for each of the three cooling periods (from 70 to 60, from 60 to 50, and from 50 to 40 F). To secure a better picture of this cooling process, plot the data of Table 1 in a diagram such as is shown in Fig. 6.

It appears from this diagram that the heat stored in the several parts of the building varies directly as the difference between the temperature under consideration and 20 F, and that it is sufficient to calculate the heat stored at some one temperature — preferably the maximum temperature under consideration. The



Figs. 1, 2, and 3—A small building designed for use in a study of intermittent heating. . . . Fig. 4—The varying temperatures in different parts of the building as the temperature of the building, or rather that of the air within the building, is lowered from 70 to 40 F

heat stored at other temperatures can then be determined graphically by means of a diagram like that of Fig. 6.

It also appears from Fig. 6 that the quantity of heat stored in the air within the building is so small that it may be neglected, and that in many cases the quantity of heat stored in the furniture in a building is so small that it may also be neglected.

Rates of Heat Loss

Having determined the quantities of heat stored in the building at successive intervals during the cooling process, and the respective differences between those quantities, which, in the present case, are 381,800 each, the next step is to calculate the rates at which the building loses heat during the respective intervals. This is done in the usual manner, and the values recorded as shown in Table 2, and for convenience of study also in Fig. 5. It appears from Fig. 5 that the rate of heat loss also varies directly as the temperature difference, and it is sufficient to calculate the heat loss for one temperature difference — preferably the maximum difference. The losses for other temperature differences may be found graphically, as shown in Fig. 5.

Having determined the successive rates of heat loss, they may be plotted in a diagram as shown in Fig. 7, in which the four ordinates represent the rates of heat loss when the respective indoor temperatures are 70, 60, 50, and 40 F. The distances x, y, and z, between the ordinates, represent the successive periods in hours required for cooling, and the successive areas of the three trapezoids represent the losses of heat during these three periods. The values of x, y, and z can then be easily calculated to be 6.6, 8.5, and 11.9 hr. Having found these values, Fig. 8 can be constructed in which the values of x, y, and z from Fig. 7 are set off to scale. From Fig. 8, it is easy to find the temperature to which the building will cool in a given time, or to find the time in which a building will cool to a given temperature.

Formula for Cooling Time

The method just described to determine the data necessary to construct the curve of Fig. 8 is simple and direct, and easy to understand, but somewhat tedious. The same data can be more easily and accurately calculated by Equation 1.

Table 1—Quantity of Heat, Btu above 20 F, Stored in Building

Indoor Air Temperature	70	60	50	40
Air in Building	13,200	10,600	7,900	5,300
Furniture	34,600	27,600	20,700	13,800
Brick Wall Above Floor	743,100	595,400	445,300	298,100
Brick Wall Below Floor	125,300	100,000	75,400	50,000
Floor	641,000	512,300	385,200	256,000
Ceiling	258,000	207,000	155,000	103,800
Roof	96,000	76,500	58,100	38,800
Total	1,911,200	1,529,400	1,147,600	765,800
Differences	381,800	381,800	381,800	

Table 2—Rate of Heat Loss, Btu per Hr, of the Building

Indoor Air Temperature	70	60	50	40
Walls	23,000	18,400	13,800	9,200
Windows	13,500	10,800	8,100	5,400
Floor	3,900	3,100	2,300	1,600
Ceiling	16,300	13,100	9,800	6,500
Air Changes 8000 cfh	7,300	5,800	4,400	2,900
Total	64,000	51,200	38,400	25,600
Differences	12,800	12,800	12,800	

$$x = \frac{h}{l} \ln \frac{t_o - t_a}{t - t_a} \quad \ldots\ldots [1]$$

where

x = the required time of cooling, hours;
h = the heat stored in the building and its contents, Btu per F;
l = the heat loss of the building in Btu per hr per F;
ln = natural logarithm
t_o = the initial indoor air temperature, F;
t = the final indoor air temperature, F;
t_a = the average outdoor air temperature, F.

To apply the formula, note from Table 1 that h, the heat stored in the building and its contents, is 38,180 Btu per F; this means that if the temperatures of the building and its contents are lowered so that the average indoor air temperature is lowered one degree, the building will lose 38,180 Btu. Similarly, if the temperatures of the building and its contents are to be raised so that the average indoor air temperature is raised one degree, 38,180 Btu must be supplied to the building by the heating system.

Note from Table 2 that l, the heat loss of the building for one degree difference between the temperature of the indoor and that of the outdoor air, is 1,280 Btu. Note also (from the preceding discussion) that the initial indoor air temperature is 70, the average outdoor air temperature is 20, and the final indoor air temperatures are, respectively, 60, 50, and 40 F. Substituting these values in Equation 1, we have successively:

$$x = \frac{38{,}180}{1{,}280} \ln \frac{70 - 20}{60 - 20} = 29.8 \times 0.223 = 6.6 \text{ hr}$$

$$x = \frac{38{,}180}{1{,}280} \ln \frac{70 - 20}{50 - 20} = 29.8 \times 0.513 = 15.3 \text{ hr}$$

$$x = \frac{38{,}180}{1{,}280} \ln \frac{70 - 20}{40 - 20} = 29.8 \times 0.916 = 27.2 \text{ hr}$$

These values check well with those previously determined and represented in Fig. 8.

Time to Heat a Building

A similar method may be used to calculate the time required to heat a building. To explain this in detail, we will apply the method described by H. Hohme in an article entitled "Temperaturverlauf beim Anheizen und Abkühlen grosser Räume" published in *Wärme*, October 17, 1931. We will use the notation relating to Equation 1 and add thereto S, the heat supplied to the building by the heating system, Btu per hr, and let x, represent the time required for heating and cooling.

Since the heat supplied to the building by the heating system in any interval of time, dx, is used to replace the heat loss by the building during that interval of time, and the excess, if any, is consumed in raising the temperature of the building and its contents, we have

$$S\,dx = l\,(t - t_a)\,dx + h\,dt \quad \ldots\ldots [2]$$

$$dx = \frac{h\,dt}{S - l(t - t_a)} \quad \ldots\ldots [3]$$

$$x = h \int \frac{dt}{S + lt_a - lt} \quad \ldots\ldots [4]$$

If we place $S + lt_a = m$, we have

$$x = h \int \frac{dt}{m - lt} \quad \ldots\ldots [5]$$

If we place $m - lt = y$, we have $dt = -\frac{dy}{l}$ and

$$x = -\frac{h}{l} \int \frac{dy}{y} = -\frac{h}{l} \ln y + C \quad \ldots\ldots [6]$$

If we replace the terms for which substitutions were made above and rearrange them, we have

$$x = -\frac{h}{l} \ln [S + l\,(t_a - t)] + C \quad \ldots\ldots [7]$$

To find the value of C, we know that $t = t_o$ when $x = 0$, and we have:

$$C = \frac{h}{l} \ln [S + l\,(t_a - t_o)] \quad \ldots\ldots [8]$$

By substituting this value of C in Equation 7 and simplifying, we have

$$x = \frac{h}{l} \ln \frac{S - l\,(t_o - t_a)}{S - l\,(t - t_a)} \quad \ldots\ldots [9]$$

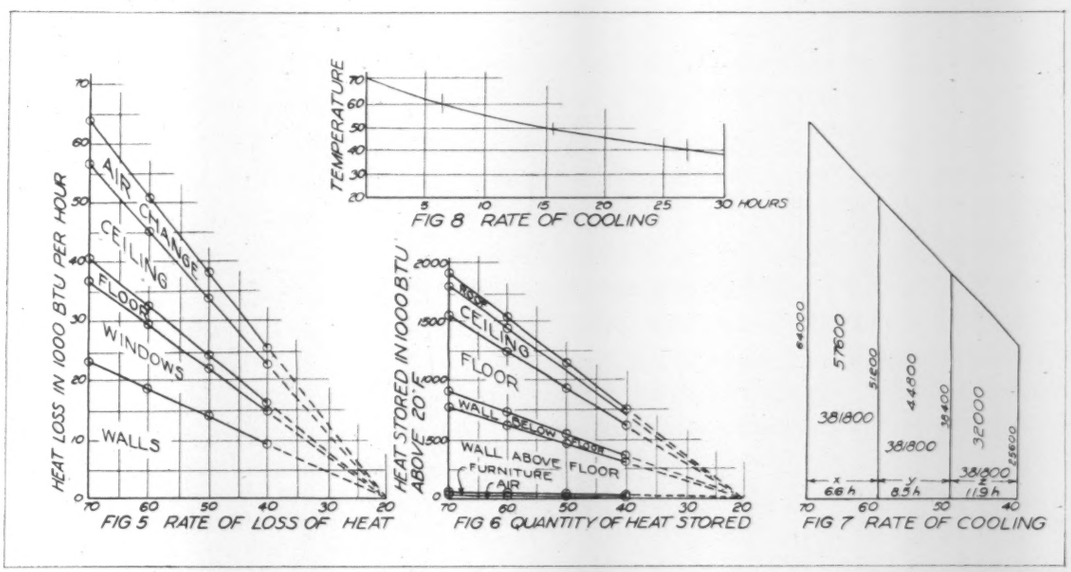


Fig. 5—Rate of loss of heat of the building when the average outdoor temperature is 20 F. . . Fig. 6—Quantity of heat, above 20 F, stored in the building and its contents for various indoor air temperatures. . . . Figs. 7 and 8—Rate of cooling of the building when the average outdoor temperature is 20 F

If the heating of the building is discontinued, the term S in Equation 9 becomes zero and that equation reduces to the form shown in Equation 1. To apply Equation 9 to calculate the time required to heat a building, it is necessary only to substitute appropriate values and to solve for x.

For example, let us assume that the building under consideration has been allowed to cool to 50, that the average outdoor air temperature is 20, that the heating system has an excess capacity of 25 per cent, *i. e.*, that its capacity is 64,000 + 16,000 or 80,000, and that it is required to find the time necessary to heat the building to 70 F. Substituting in Equation 9 we have

$$x = \frac{38{,}180}{1{,}280} \ln \frac{80{,}000 - 1280\ (50 - 20)}{80{,}000 - 1280\ (70 - 20)} = 29.8 \ln 2.6 = 28.5 \text{ hr}$$

However, if the heating system had an excess capacity of 100 per cent, *i. e.*, 128,000 Btu, the required time to heat from 50 to 70 would be only 10 hr.

Equations 1 and 9 may be used to construct diagrams from which data regarding cooling-down and heating-up may easily be obtained for the particular building for which the diagram was designed. For example, for the small building of Fig. 1, Fig. 9 shows that when the average outdoor temperature is 30, the building will cool from 70 to 50 in about 21 hr, and that the same degree of cooling would require about 33 hr if the average outdoor temperature were 40. For outdoor temperatures differing from those shown in Fig. 9, it is comparatively easy to interpolate with sufficient accuracy.

Similarly, it appears from Fig. 10 that with a heating system having an excess capacity of 50 per cent, or a total capacity of 96,000 Btu per hr, it will require about 14 hr to heat from 50 to 70 when the average outdoor temperature is 30, and about 18 hr when the average outdoor temperature is 20.

Continuous or Intermittent Heating?

In all heating installations it is important to know whether it is more economical to heat continuously—24 hr every day—or intermittently, say, 20 or 16 or 12 hr a day. This problem can be studied nicely with the aid of the diagrams of Figs. 9 and 10. For example, let us assume that the small building under consideration is a school, in which a 70 F indoor temperature is to be maintained from 8 a.m. to 5 p.m., and that heating can be discontinued at 5 p.m. Let us find when heating must be resumed if the average outdoor temperature is 20, 30, or 40 and if the available heating system has a capacity of 96,000 Btu per hr. To find the answer, construct a diagram as shown in Fig. 11 and apply the $t_a = 20$ or 30 or 40 curve from Fig. 9, so as to begin at 5 p.m. and the $t_a = 20$ or 30 or 40 curve from Fig. 10, so as to end at 8 a.m. The intersections of these curves show the times—about 10:45, or 12:45, or 2:45—when heating must be resumed in the three cases respectively. They also show that the building will have cooled to about 61 F in each of the three cases.

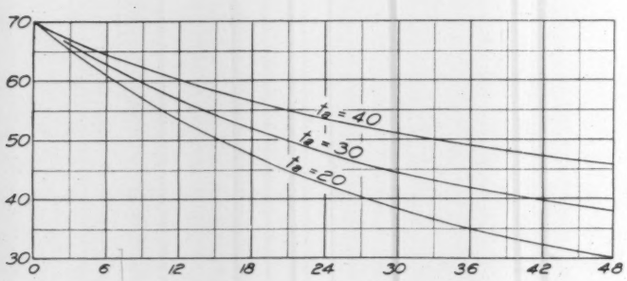


Fig. 9—Cooling-down periods of the building, in hours, for various outdoor temperatures

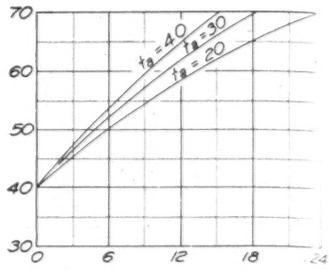


Fig. 10—Heating-up periods of the building, in hours, for various outdoor temperatures, and for a heating system supplying 96,000 Btu per hr

To determine the economy of interrupting the heating, when the outdoor temperature is 20 F, for the 5.75 hr period, from 5 p. m. until 10:45 p. m., note that since the heat loss of the building is 1,280 Btu per hr per F, and since the difference between indoor and outdoor temperatures is assumed 70 — 20, or 50 deg, the heat which would have been supplied to the building from 5 p. m. to 8 a. m. for continuous heating would have been 15 × 50 × 1,280 or 960,000 Btu. Note also that with intermittent heating no heat would have been supplied from 5 p. m. until 10:45 p. m., and that from 10:45 p. m. until 8 a. m., for a period of 9.25 hr, heat would have been supplied at the rate of 96,000 Btu per hr, or a total of 888,000 Btu, thereby effecting an economy of 960,000 — 888,000, or 72,000 Btu per day. If heat costs 50 cents per 1,000,000 Btu, a daily saving of 72,000 Btu is equivalent to a daily saving of 3.6 cents.

Similar calculations for the two cases when the outdoor temperature is 30 or 40 show that, in those cases, the saving is also 72,000 Btu per day. It may seem strange that the saving should be practically the same in each of the three cases; that this is true is shown by the following reasoning:

As the building cools below 70, the heat loss is reduced in proportion to the cooling. The total heat saved, in each of the three cases, is equal to the area of one of the three triangles $a\ b\ c$ of Fig. 11—in degree-hours—multiplied by 1,280, the heat loss of the building per degree-hour. Since the three triangles have a common base $a\ b$, and since their altitudes are about equal and their sides very similar, their areas are approximately equal and, therefore, the saving due to intermittent heating is approximately equal.

However, although the absolute saving is practically independent of the outdoor temperature, the percentage saving increases as the outdoor temperature rises, and is about 4.7, 5.9, and 7.8 per cent when the outdoor temperature is 20, 30, and 40 F, respectively.

If it costs more to produce heat at the higher rate necessary during the heating-up period of intermittent heating than it costs at the lower rate sufficient for continuous heating, the resulting increased cost must be deducted from the saving calculated above to secure the net saving resulting from intermittent heating.

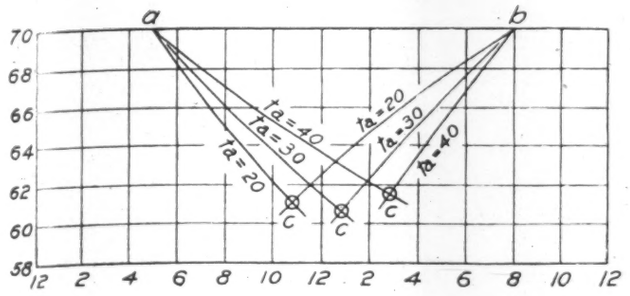


Fig. 11—Graphical method of determining the time when heating must be resumed in a building which is heated intermittently

For example, in a small heating system the efficiency may be reduced when the system is operated above its normal capacity, and in a large central heating system, extracted steam may be available in sufficient quantity to carry the steady heating load of continuous heating, whereas it may be necessary to use high pressure steam during the heating-up period of intermittent heating, and this high pressure steam may cost more than the extracted steam.

Heat Loss When Building Cools 1 F

The two principal difficulties with the method described above for calculating the time required for cooling and heating are first, to determine the average outdoor temperature during the cooling and heating period, and, second, to determine the value of h in Equations 1 and 9; that is, the quantity of heat which the building and its contents lose when the average indoor air temperature is lowered one degree, or the quantity of heat which must be supplied to the building and its contents when the average indoor air temperature is raised one degree. This quantity of heat can be determined fairly accurately if it is assumed that the temperature gradient through the walls, floor, ceiling, and roof can be represented by a straight line during the cooling and heating periods, as it can be when the heating process has attained a steady state.

This was done for the construction of Fig. 4 (*A*, *B*, *C*, and *D*). A study of this figure shows that when the building—*i.e.*, the indoor air—cools one degree, the mass of the building cools less than one degree. For example, it appears from Fig. 4 (*A* and *D*) that when the temperature of the indoor air lowers from 70 to 40, or 30 deg, the average temperatures of the elements of the building lower as follows: Exterior walls above floor, from 41.2 to 28.5, or 12.7 or 42 per cent of 30 deg. Ceiling, from 57.5 to 35, or 22.5 or 75 per cent. Roof, from 29.4 to 23.8, or 5.6 or 19 per cent. Floor, from 66.5 to 38.6 or 27.9 or 93 per cent. Exterior wall below floor, from 38.3 to 27.3, or 11.0 or 37 per cent.

Consequently, in calculating the heat dissipated by a building while it is cooling one degree, it is necessary to use only certain percentages of the several parts of the building, as indicated above. This is illustrated in the calculations shown in Table 3, which apply to the building under consideration.

Interior walls, and floors between successive stories, dissipate their heat during the cooling period to the exterior walls and/or to the roof, partly by air convection currents and partly by radiation. The percentages of their masses which should be used in heat loss calculations will vary from building to building. Unless definite percentages have been calculated, it may be satisfactory to assume 80 per cent for these elements.

In addition to determining the percentages of the weights of the several parts of the building to be used in calculating the value of h, it is also necessary to take into account the time lag that exists when the temperature of the building changes. For example, Fig. 12 shows how the temperature gradient must be represented by a series of curved lines as the indoor temperature gradually falls from 70 to 40 F, and that consequently the 42 per cent shown above is slightly too high. When the indoor temperature has fallen to 40, as shown in Fig. 12, and is then held at 40, the temperature gradient will change gradually and approach closer and closer to the straight line shown in Fig. 4 *D*. During this period the excess heat stored in the central portion of the wall is gradually conducted toward the outer surfaces. A similar condition exists during the heating-up period.

Table 3

Part of Building	Total Weight	Per Cent Used	Weight	Specific Heat	Heat Stored
Exterior walls above floor	175,000	42	73,600	0.20	14,720
Ceiling	34,500	75	25,800	0.20	5,160
Roof	19,600	19	3,700	0.52	1,930
Floor	69,000	93	64,200	0.20	12,840
Exterior walls below floor	34,300	37	12,700	0.20	2,540
Air	1,200	100	1,200	0.24	290
Furniture	2,000	90	1,800	0.52	940
Total					38,420

With interior walls and with floors in buildings of several stories, the time lag is of still greater influence. The estimate of 80 per cent suggested above for such walls and floors is only approximate. Experimental determinations of the influence of the heat storage capacity of interior walls and floors on the rates of cooling-down and of heating-up of buildings will be of great value and are necessary to provide data for more accurate calculations than are possible now.

The flow of heat in and through walls and floors has an important bearing on time lag and deserves careful study.

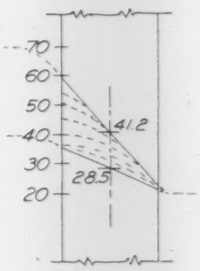


Fig. 12 — Variation of the temperature gradient in the outside wall of the building shown in Figs. 1, 2, and 3, as the building cools from 70 to 40 F

Check Charts for Air Conditioning Calculations

By **J. M. Dean***

A GENERAL introduction to this set of charts was given in the May issue of H.P.&A.C.

Charts 5 and 6 are published on the next page. Chart 5 is for checking sensible and latent heat of occupants, heat load due to lights, fan motor and duct gains, and converting cfm and cfh. Chart 6 is for determining cfm requirements.

*Air Conditioning Engineer, Des Moines Electric Light Co. All rights reserved.

[See preceding page]

Chart No. 5

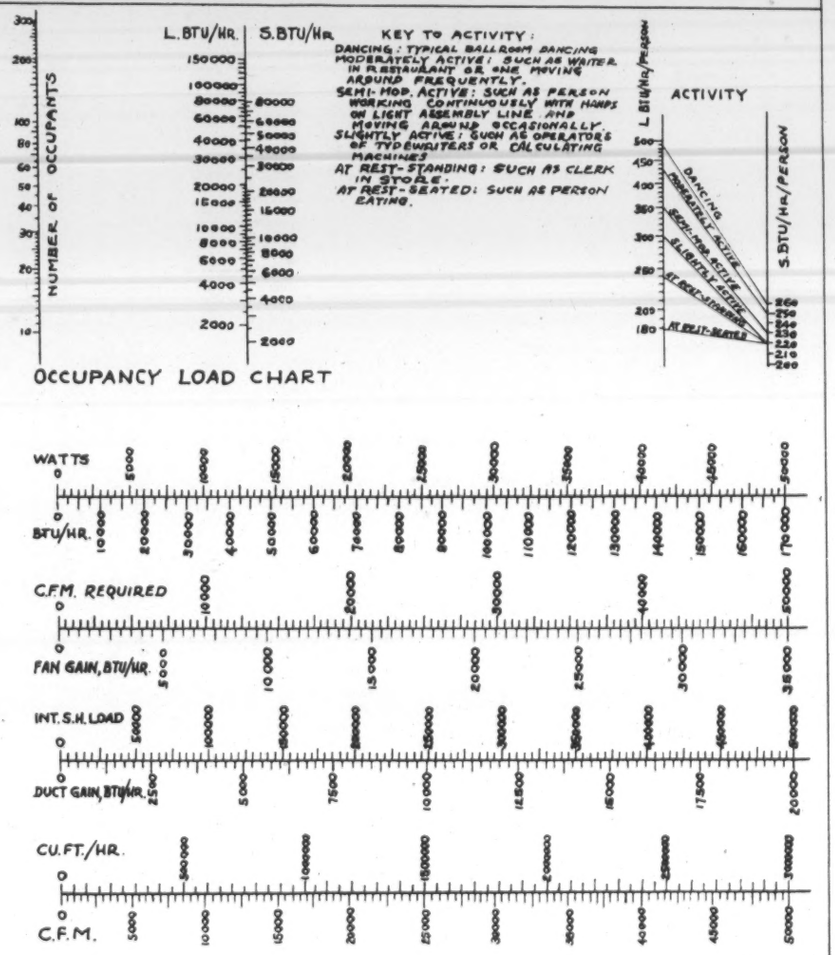
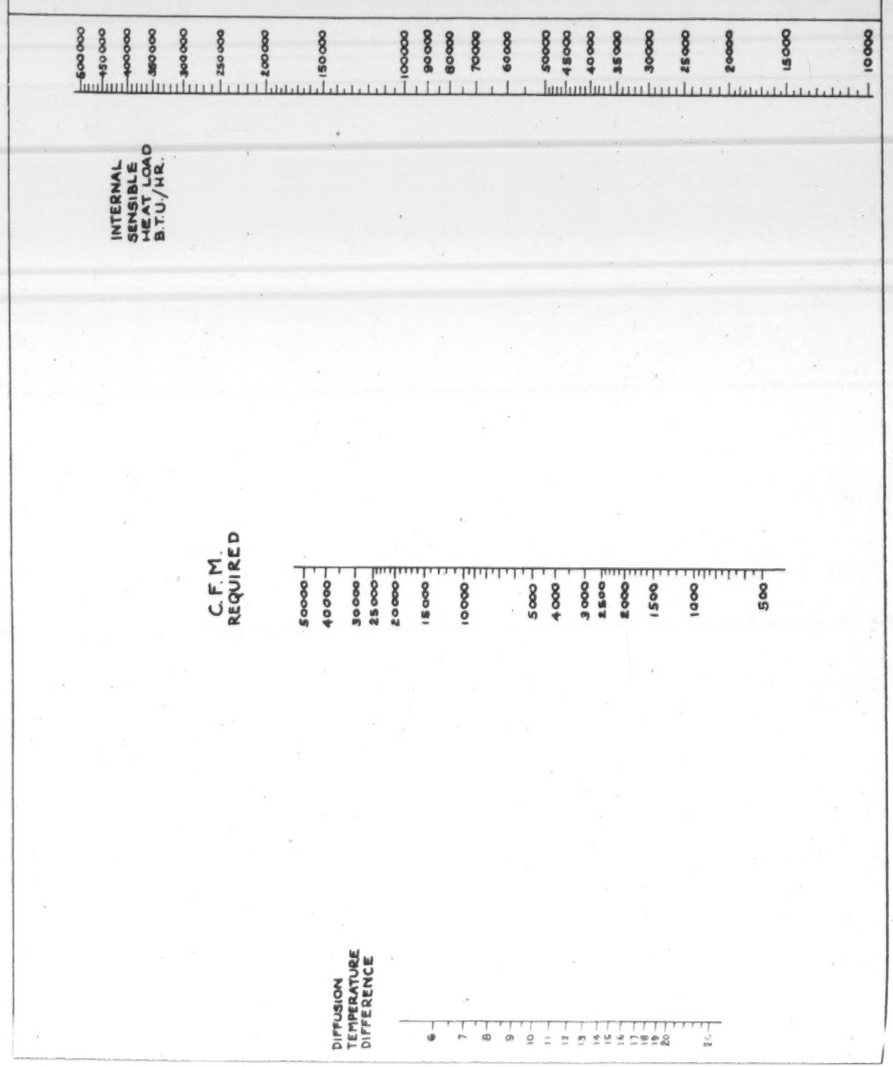


Chart No. 6



Figuring Solar Heat Gains of Buildings

By William Goodman*

IN the May and June issues, tables were presented—and their use explained—for computing the solar heat gains through walls, roofs, and windows. Last month, discussion was started of the theory underlying the subject of solar heat gains and the methods used in preparing the tables previously published. This discussion is continued this month.

Maximum Solar Energy Falling on a Black Surface

Measurements of the amount of solar energy (I_p)† falling on a black surface perpendicular to the rays of the sun have been made by the weather bureau and by the A.S.H.V.E. laboratory. Computations should not, of course, be based on the very maximum recorded as such a maximum occurs rarely and when it does, is of only short duration. A curve of values recommended for actual use is given on p. 150 of the *Heating, Ventilating, Air Conditioning Guide*, 1938. The values used in the preparation of Tables 1-8 were read from this curve and are given in Table 10.

The amount of heat falling on a black perpendicular surface varies with the time of day because of the variation in the length of the path of the rays of the sun through the atmosphere surrounding the earth. The amount of solar energy reaching the surface of the earth is less than the amount reaching the stratosphere. As the rays travel through the atmosphere surrounding the earth, much of their energy is absorbed by the dust and water vapor suspended in the atmosphere. The amount absorbed—or the *atmospheric depletion*, as it is known—depends upon the length of the path that the rays of the sun follow to reach a particular point on the surface of the earth. As can be seen in Fig. 5, the length of this path is shortest at noon and longest in early morning and late afternoon.

Table 10—Solar Heat Falling on a Surface Normal to the Sun, Btu per Hr

Time A.M.	I_p	Time P.M.
6	60	6
7	175	5
8	255	4
9	285	3
10	302	2
11	307	1
12	310	12

Although the values of I_p listed in Table 10 were measured in a latitude of 40 deg north, these values were also used in computing the solar heat gain for latitudes down to zero degrees. As I_p is a quantity that varies widely and erratically, the judgment of the experimenter enters to some extent in selecting the values to be recommended for design purposes. In view of this, it is probable that the use of the values in Table 10 is also satisfactory for the lower latitudes.

Theory of Solar Heat Gains Through Bare Glass

The values of I_u obtained by means of Equation 3 are not the quantities of solar heat transmitted into the room: these values are only the amount of solar heat falling on the outside of each square foot of glass. The amount of heat transmitted through a glass surface into a room is always less than the amount impinging on the exterior side of the glass.

Part of the light falling on a glass surface is reflected, part is transmitted through the glass, and part is absorbed by the glass itself. The amount transmitted depends upon the angle of incidence. The larger the angle of incidence, the smaller will be the amount of light transmitted, and the larger the amount reflected. The energy absorbed by the glass itself is always small regardless of the angle of incidence.

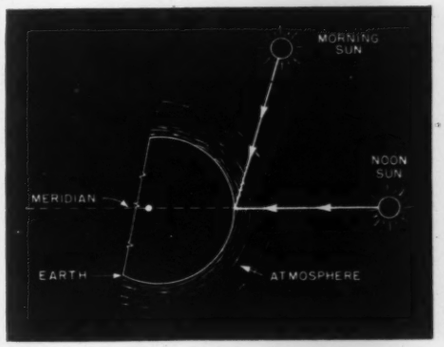


Fig. 5—The length of the path of the rays of the sun through the atmosphere of the earth is longer in the early morning or late afternoon than at noon

These statements are illustrated in Fig. 6 for single and for double glass (two separate, parallel thicknesses of glass). Notice that the maximum amount of energy is transmitted when the ray of light is perpendicular to the glass surface—that is, the angle of incidence is zero. Particularly interesting is the fact that the amount of energy transmitted through glass is practically constant and equal to the maximum amount for all angles of incidence up to about 45 deg. For larger angles, the amount transmitted falls quite rapidly, whereas the amount reflected increases rapidly. These curves also show that the amount of energy absorbed by the glass itself is always a small percentage of the total, regardless of the angle of incidence.

The curves in Fig. 6 were computed by means of the following formulas, whose derivations are given in Appendices 3, 4, 5, and 6.

Single Glass

Per cent transmitted

$$P_{T1} = (1 - P_F)^2 e^{-kL} \quad [7]$$

Per cent absorbed

$$P_{A1} = (1 - P_F)\ (1 - e^{-kL}) \quad [8]$$

Double Glass

Per cent transmitted

$$P_{T2} = P^2_{T1} \quad [9]$$

Per cent absorbed

$$P_{A2} = (1 + P_{T1})\ P_{A1} \quad [10]$$

In these equations P_F is the percentage of the incident energy which is reflected from the outside surface of the glass; the balance being transmitted through the out-

*The Trane Co. Member of Board of Consulting and Contributing Editors.

Part 4. Part 1 was published in Heating, Piping and Air Conditioning, May, 1938, pp. 315 318; Part 2 in June, pp. 391-394; and Part 3 in July, pp. 445-447.

†See list of symbols on p. 446, July issue.

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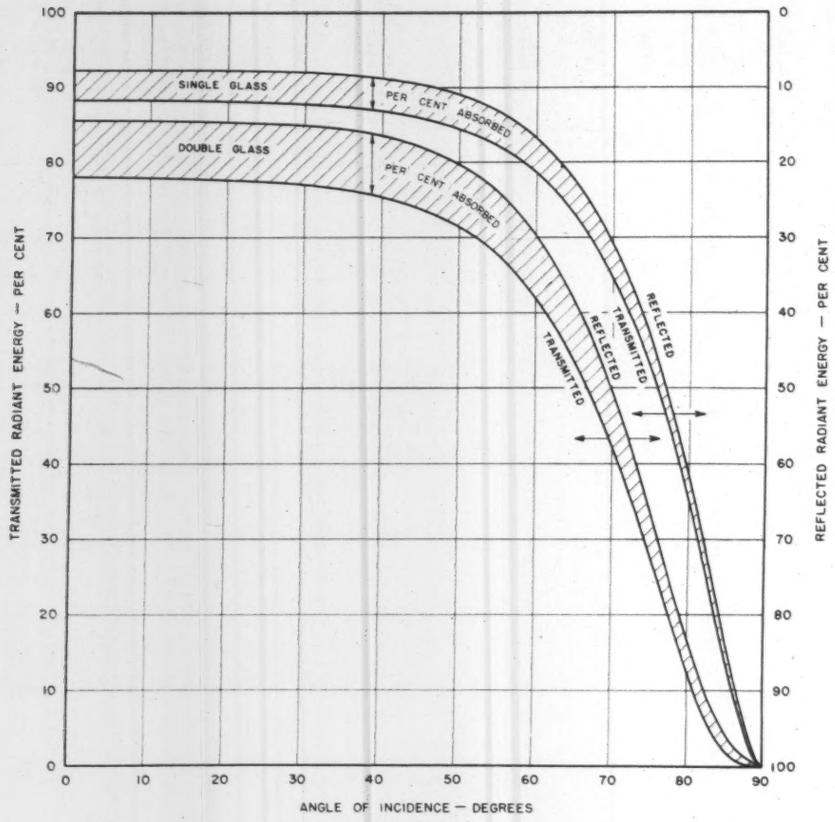


Fig. 6—Variation in the solar energy transmitted and reflected by glass with changing angles of incidence

side surface. P_F was computed by means of Fresnel's formula, which is given in Appendix 7. The value of P_F varies with the angle of incidence.

In [7] and [8], k is an experimentally determined constant; its numerical value is the same in both of these equations. If the amount of light transmitted through single glass at zero angle of incidence is known, the value of k can be determined, and all other points on the four curves of Fig. 6 can then be computed by means of Equations 7 to 10. Shaver[4] reports that at zero angle of incidence a single pane of glass transmits 88 per cent and reflects 8 per cent of the incident light. This leaves 4 per cent as the amount absorbed by the glass. This is in substantial agreement with the following results reported in an A.S.H.V.E. test.[5]

Per cent reflected and absorbed	Per cent transmitted
10.8	89.2
11.2	88.8

In order to verify the validity of [7], the curve in Fig. 6 of the energy transmitted through single glass was replotted in Fig. 7. On this same figure are plotted the actual points obtained in the A.S.H.V.E. test.[5] Most of the experimental points fall along the theoretical curve. The points for large angles of incidence deviate more from the curve than do the points for small angles of incidence. At the larger angles, the per cent transmitted varies so much for even a small change in the angle of incidence that any slight error in the test would throw the point considerably off the curve. Considering all factors, the agreement between the test points and the theoretical curve may be considered as fairly satisfactory. As an additional check, it is interesting to note that Shaver[4] also reports that 77 per cent of the incident energy is transmitted through double glass at zero angle of incidence. The A.S.H.V.E. test[5] reports a value of 72.2 per cent transmitted. (27.8 per cent reflected and absorbed.) Figured by means of Equation 9, 78.0 per cent is transmitted through double glass (see Appendix 7). This is in good agreement with Shaver's value, but somewhat higher

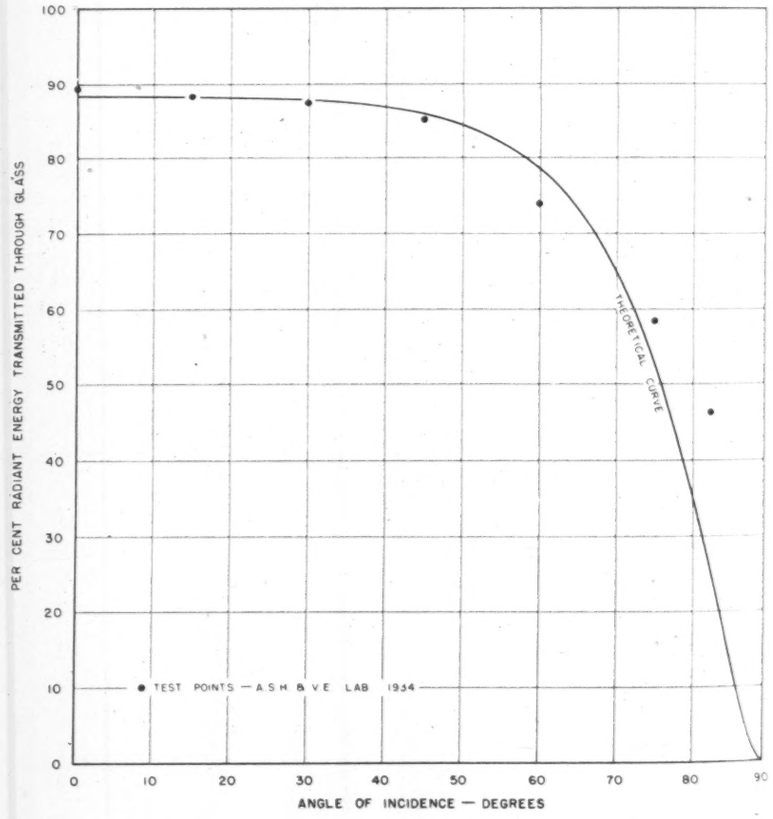


Fig. 7—Comparison between theoretical curve for solar heat transmitted through single glass and A.S.H.V.E. test points

[4]Windows and Their Relation to Air Conditioning Problems, W. W. Shaver, *Refrigerating Engineering*, September, 1933, Table 2, p. 134.

[5]Studies of Solar Radiation Through Bare and Shaded Windows, Houghten, Gutberlet, and Blackshaw. *A. S. H. V. E. Journal Section*, Heating, Piping and Air Conditioning, February, 1934, Table 4, p. 73.

Table 11—Fraction of Solar Heat Transmitted Through Single and Double Glass

cos i	Single Glass P_{H1}	Double Glass P_{H2}	cos i	Single Glass P_{H1}	Double Glass P_{H2}	cos i	Single Glass P_{H1}	Double Glass P_{H2}
0	0	0	0.35	0.7060	0.4645	0.70	0.9035	0.7750
0.01	0.0075	0.0005	0.36	0.7175	0.4800	0.71	0.9050	0.7780
0.02	0.0175	0.0008	0.37	0.7280	0.4955	0.72	0.9070	0.7810
0.03	0.0320	0.0010	0.38	0.7390	0.5105	0.73	0.9080	0.7835
0.04	0.0515	0.0025	0.39	0.7490	0.5250	0.74	0.9090	0.7865
0.05	0.0730	0.0045	0.40	0.7585	0.5385	0.75	0.9105	0.7885
0.06	0.0960	0.0075	0.41	0.7680	0.5525	0.76	0.9115	0.7915
0.07	0.1190	0.0120	0.42	0.7770	0.5660	0.77	0.9125	0.7935
0.08	0.1430	0.0180	0.43	0.7850	0.5785	0.78	0.9130	0.7960
0.09	0.1690	0.0255	0.44	0.7930	0.5910	0.79	0.9145	0.7975
0.10	0.1950	0.0350	0.45	0.8010	0.6025	0.80	0.9150	0.7995
0.11	0.2230	0.0450	0.46	0.8076	0.6140	0.81	0.9160	0.8010
0.12	0.2570	0.0570	0.47	0.8150	0.6250	0.82	0.9170	0.8025
0.13	0.2780	0.0700	0.48	0.8220	0.6365	0.83	0.9175	0.8040
0.14	0.3050	0.0840	0.49	0.8280	0.6470	0.84	0.9185	0.8050
0.15	0.3300	0.0995	0.50	0.8340	0.6570	0.85	0.9190	0.8065
0.16	0.3590	0.1155	0.51	0.8400	0.6665	0.86	0.9200	0.8075
0.17	0.3780	0.1325	0.52	0.8455	0.6750	0.87	0.9205	0.8085
0.18	0.4025	0.1500	0.53	0.8500	0.6835	0.88	0.9210	0.8095
0.19	0.4260	0.1680	0.54	0.8550	0.6915	0.89	0.9220	0.8105
0.20	0.4500	0.1865	0.55	0.8595	0.6990	0.90	0.9225	0.8115
0.21	0.4720	0.2050	0.56	0.8635	0.7065	0.91	0.9225	0.8120
0.22	0.4925	0.2245	0.57	0.8675	0.7125	0.92	0.9226	0.8130
0.23	0.5125	0.2430	0.58	0.8720	0.7190	0.93	0.9227	0.8135
0.24	0.5330	0.2625	0.59	0.8751	0.7250	0.94	0.9228	0.8140
0.25	0.5525	0.2825	0.60	0.8785	0.7310	0.95	0.9229	0.8140
0.26	0.5710	0.3025	0.61	0.8820	0.7370	0.96	0.9230	0.8145
0.27	0.5890	0.3225	0.62	0.8850	0.7420	0.97	0.9230	0.8145
0.28	0.6050	0.3420	0.63	0.8880	0.7470	0.98	0.9231	0.8150
0.29	0.6200	0.3605	0.64	0.8905	0.7515	0.99	0.9231	0.8150
0.30	0.6375	0.3785	0.65	0.8930	0.7565	1.00	0.9232	0.8155
0.31	0.6525	0.3970	0.66	0.8950	0.7600			
0.32	0.6675	0.4145	0.67	0.8975	0.7645			
0.33	0.6830	0.4320	0.68	0.8995	0.7680			
0.34	0.6930	0.4480	0.69	0.9020	0.7715			

than the value which was reported in the A. S. H. V. E. test.

The energy absorbed by the glass appears as heat in the glass itself and raises the temperature of the glass. The amount transmitted through the glass passes through as radiant energy and is not transformed into heat until it is intercepted by the opaque surfaces inside the room.

Whether the heat absorbed by the glass will be partly or wholly transferred to the room air depends upon the temperature to which the glass is heated by the absorbed energy. If the temperature of the glass is raised to a point that is higher than the temperature of the outdoor air, then part of the absorbed energy will be transferred to the outdoor air. The available experimental evidence on this point is too meager to permit of a definite answer. Inasmuch as the curves of Fig. 6 show that the amount of heat absorbed by clean glass is always a small percentage of the total, the temperature of the glass was assumed to be lower than the temperature of the outdoor air. In the preparation of Tables 1 to 8, the small amount of energy absorbed by single glass was assumed ultimately to be transferred to the room, in addition to the large amount transmitted directly through the glass as radiant energy.

The tables of solar heat gains through double glass were prepared on the assumption that only the solar heat absorbed by the second thickness of glass was transferred to the room. It does not seem reasonable to suppose that the heat absorbed by the first pane will also be transferred to the room. In the case of a single thickness of glass alone, the inside surface is in contact with the room air and is thus kept cool. However, in the case of the double thickness, the inside surface of the outer pane is in contact with the air space between the two panes. For this reason, it is quite possible in this case that the temperature of the first pane may be raised to such a point that the absorbed heat will be transferred back to the outdoor air. The amount absorbed by the second thickness alone is slightly less than the amount absorbed by the first thickness alone.

No allowance was made in the tables for a small amount of diffuse radiation from the open sky. This diffuse radiation from the open sky into rooms is always present on clear bright days, even though the windows of the room face in a direction opposite to the one from which the sun is shining. The method of computing the heat gain through windows outlined here is sufficiently safe to permit diffuse radiation to be ignored.

In order to compute Tables 1-8, values of I_u were first computed by means of Equation 3. Values of the percentage of energy transmitted and absorbed by single and double glass were read from Table 11 for various angles of incidence. The values in Table 11 were computed as described in Appendix 7. For single glass, the values in the column headed P_{H1} were multiplied by I_u in order to find the solar heat transmitted through the glass. The values of P_{H1} were found by the following formula:

$$P_{H1} = P_{T1} + P_{A1} \quad [11]$$

For double glass, the values in the column headed P_{H2} were multiplied by I_u in order to find the solar heat transmitted through the glass. The values of P_{H2} were found by the following formula:

$$P_{H2} = P_{T2} + (P_{A2} - P_{A1}) \quad [12]$$

Whether heat will flow from the outdoor air to the outside surface of a window, and then in turn be transferred to the room, depends upon how much the temperature of the glass is raised by the solar heat absorbed. Inasmuch as the amount of radiant energy absorbed by clean glass is small, it is questionable whether the temperature of the glass can be increased much. For the present, therefore, it seems best to assume that the normal conduction of heat through glass is unaffected by the solar heat absorbed. For this reason, the conduction heat gain through the glass should always be figured in addition to the solar heat gain. This is discussed further in Appendix 8.

Example 5 illustrates the procedure for computing the values in Tables 1 to 8.

Example 5: Compute the solar heat gain of one square foot of single glass facing east in a latitude of 45 deg. The time is 9 a. m.

Solution:

From Table 9, $\cos i = 0.6646$

From Table 10, $I_D = 285$ Btu

$$I_u = I_D \cos i \quad [3]$$

$$= 285 \times 0.6646$$

$$= 189.4 \text{ Btu}$$

From Table 11, for single glass and for $\cos i = 0.6646$, $P_{H1} = 0.895$

Solar heat transmitted through glass $= 0.895 \times 189.4 = 169.5$ Btu per hr.

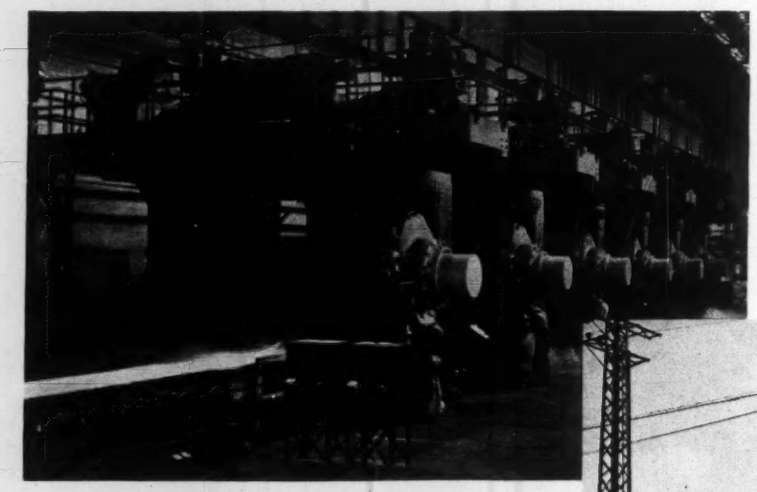
[To be continued]

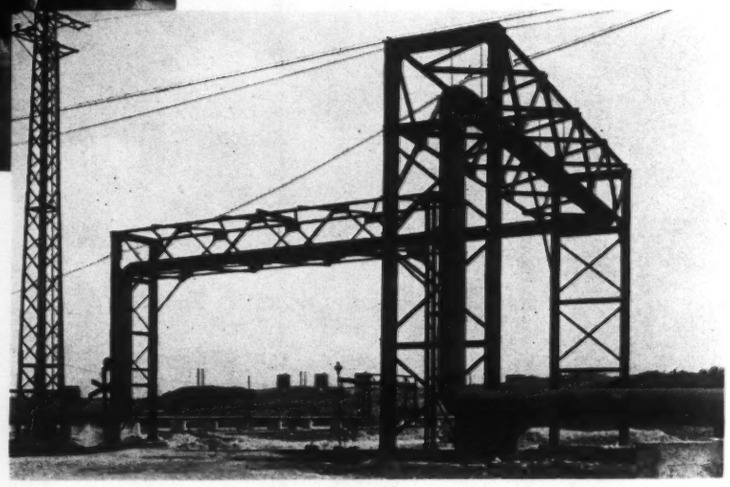
Increased production in the caramel department of "100 per cent during more than one-half of the days of the average summer" since installation of air conditioning has been reported by the Nutrine Candy Co., Chicago. A 50 per cent increase in production of chocolate dipping of cream and cherry centers was accomplished by speeding up, through air conditioning, the center cooling and setting.

Long Pipe Line Ties In Steam Plants, Supplies New Continuous Strip Mill

By James M. Main*

The factors leading to the construction of the long welded steam line supplying Republic's new continuous strip mill at Cleveland are discussed, and the details of the piping are described





***Above*—The six stand finishing train in the hot mill. Strip steel is rolled out of the last stand at a speed of over 2100 fpm . . . . . *Right*—The steam line supplying the mill crosses railroad tracks and driveways on six 22 ft cross-overs. Traps and valves are housed in steel cabinets similar to the one in the foreground**

THE location of Republic Steel Corp.'s new 98 in. continuous strip mill offered the possibility of either constructing a steam generating plant at the mill site or installing a pipe line of the proper size to supply the strip mill requirements from Republic's nearby Corrigan, McKinney plant, which is about two miles below the strip mill site on the Cuyahoga river.

The new continuous strip mill, a colossus in an industry where machinery and equipment are commonly gigantic, is the largest and fastest mill of its kind ever built. It can roll steel in all finished widths from 30 to 94 in. and all finished thicknesses from 18 ga strip to half-inch plate. Maximum delivery speed is 2121 fpm. The plant has a nominal rated capacity of 70,000 gross tons per month.

The steam demand required a maximum of about 100,000 lb per hr of steam at 125 lb per sq in. pressure. Steam is used for the operation of a batch pickler, continuous picklers, miscellaneous industrial heating applications and for blowing scale from strip. The summer load averages approximately 35,000 lb per hr.

Why Pipe Line Instead of New Steam Plant?

In the determination of the relative merits of the two possible methods of supplying the steam demands, consideration was given to the following: The steam plant would have a higher first cost, a higher maintenance cost and have greater efficiency. The steam transmission line would require no standby equipment. In addition, certain factors affecting the heat balance and availability of fuel of the Corrigan, McKinney plant were considered.

The power generating system at the Corrigan, McKinney plant consists of:

(1) A main power station on the west bank of the river. Fuel here is blast furnace gas with coal and oil auxiliary. Capacity is 12,345 boiler hp, 750,000 lb per hr of steam. Steam pressure is 250 lb per sq in., 125 F superheat.

(2) A steel plant boiler house on the east bank. Fuel here is coke breeze and coal with auxiliary, coke oven gas. Capacity is 4938 boiler hp, 210,000 lb per hr of steam. Steam pressure is 175 lb per sq in., 75 F superheat.

(3) A waste heat boiler house, also on the east side. Fuel here is waste heat from the open hearth furnaces. Capacity is 5000 boiler hp, 150,000 lb per hr of steam. Steam pressure is 175 lb per sq in., 75 F superheat.

The desirability of the interchange of steam and fuel requirements between these plants to tie in with the variation in the general plant operation and corresponding fuel availability necessitated a steam tie-in between the main power station on the west side of the river with the steel plant boiler house on the east side. In addition to improving the balance of the general fuel and power requirements of the Corrigan, McKinney plant, it would make available, under normal plant operating conditions, sufficient steam to supply the strip mill without the addition of new steam generating equipment.

The evaluation of the above factors, together with the

*Engineer, Republic Steel Corp.

availability of the necessary right-of-way and the necessity of structures and supports for other lines not at present installed determined in favor of the steam transmission line.

Details of Piping Construction

The right-of-way, occupying in most part the old Ohio canal lands, roughly parallels the east bank of the Cuyahoga river and is used by the River Terminals Railroad Co. for track from the Corrigan, McKinney plant to the strip mill site.

The steam transmission line consists of three sections, which together form an odd-shaped, elongated "Y." The three sections are:

(1) A line from the west side power station to a junction point on the east bank of the river near Clark avenue and Independence road.

(2) A line from the steel plant boiler house to the same junction point. (This line can be used to augment the steam supply from the west side power station to the east side plants.)

(3) A line from the junction point to the strip mill.

The first line is a 10 in. line, 1960 ft long (developed length). It is supported in part by pipe pole structure. It crosses the Cuyahoga river under the sidewalk of the Clark avenue bridge, which is about 100 ft above the river.

The line is of welded construction, of 10 in. standard steel pipe. Expansion is provided for by four 10 in. offset expansion U bends. It is designed to carry a maximum flow of 150,000 lb per hr of steam. Initial pressure is 240 lb per sq in., 125 F superheat.

The line from the steel plant boiler house to the junction point is constructed of 16 in. O. D. x ⅜ in. wall pipe. It is of welded construction, supported in the most part on existing pipe racks. This line is designed for a maximum flow of 130,000 lb per hr of steam, initial pressure 165 lb per sq in., 70 F superheat.

The third line, from the junction point of the 10 in. and 16 in. lines to the strip mill, is designed for a maximum flow of 100,000 lb per hr of steam. It consists of 5860 lineal feet of 13⅜ in. O. D. x 0.43 in. wall and the remainder (2435 lineal feet) of 12 in. x 0.375 in. wall pipe. Initial steam pressure is 160 lb per sq in.

The entire line is of welded construction with Series 30 forged steel flanges, extra heavy welding ells, cast steel valves for 300 lb per sq in. working pressure. In the case of the 13⅜ in. O. D. pipe, 14 in. x 23 in. Series 30 forged steel flanges bored for 13⅜ in. O. D. pipe were used, and elbows were fabricated by welding.

For the most part the line is supported on reinforced concrete piers, spaced on about 25 ft centers. They are of sufficient size to carry an additional 30 in. line.

Expansion is provided for by the use of offset U expansion bends and cross-overs. Expansion bends were shipped in three parts and welded together in the field.

Pipe Supports

The pipe supports on the concrete piers consist of standard cast iron rollers on steel shafts supported in chairs of welded plate construction. They are secured to the piers with anchor bolts. Where required, to allow for lateral movement of the line at the expansion bends, chairs and shafts are lengthened.

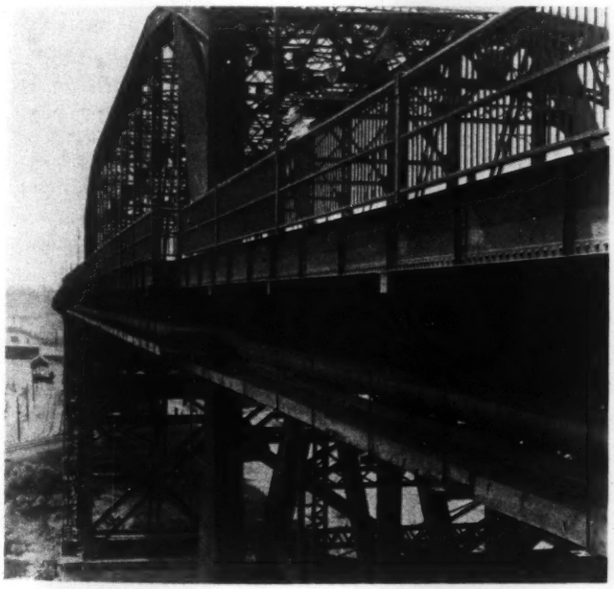
The supports on overhead structures consist of two rod swing hangers attached to pipe clamps on the pipe.

Anchors and alignment guides are of welded construction, made up of structural shapes and plates. Guide plates and anchor lugs are welded to the pipe.

The overhead line is suspended from structural bends consisting of two 6 in. x 8 in. well truss steel columns with double channel members across the top. Poles are bolted at the bottom to galvanized angles cast in pocket in concrete piers.

Altogether there are six railroad and highway crossovers, 22 ft above ground. They consist of structural box trusses with the pipe suspended from cross members which are supported on upper chords.

The line crosses the Cuyahoga river under the sidewalk of the Clark avenue bridge



Trap and Valve Cabinets

Condensation pockets, drained by forged steel inverted bucket traps, are situated at all low points and at intervals of about 500 ft. Traps and valves are housed in steel cabinets with locks to prevent tampering. Cabinets are attached to drip pockets in accessible positions, alleviating the necessity of swing connections to provide for the expansion.

The line is insulated with double standard thickness 85 per cent magnesia sectional covering with waterproof jacket. Standard protection saddles are provided at all roller supports.

All welding was done by means of the electric arc. Welds are 60 deg V butt type, laid in three beads.

The line was put in service November 15, 1937, and has been in continuous use since that time. The maximum flow up to the present time has been 75,000 lb per hour.

The steam line was fabricated and erected according to the design and under the supervision of the engineering department of the Republic Steel Corp., Corrigan, McKinney Div.

Research Committee Studies Grounding of Electrical Circuits on Piping

By H. S. Warren*

THE protective grounding of electric circuits and apparatus in homes has been the approved standard practice for many years and is essential to the safe use of electricity under the conditions generally prevailing in this country. This grounding practice is covered by mandatory rules in the National Electrical Code and the National Electrical Safety Code. It has become almost universal, not only in the U. S. A., but also in other countries where there is a high electrical development.

By "protective grounding" is meant the electrical connection of circuits or equipment to the earth. The object of such grounding is to limit the voltage which a person touching the circuit may accidentally sustain. It is very important that the circuit be grounded in the most effective way practicable. If a house is piped for water or gas, the circuit should be grounded on such pipe, preferably the water pipe, for a water pipe, if connected to a continuous metallic water supply system, affords by far the largest available area of contact with the conducting material of the earth. If, however, the house contains no such pipes or other large grounded metal body, the circuit should be grounded through a driven pipe, or rod, or some other buried electrode providing the best practicable contact with earth.

While necessity for protective grounding is recognized by electrical engineers, some pipe-owning companies, who are not usually very familiar with electrical problems, have objected to the use of their pipes by others (as it seemed to them), for a purpose they did not fully understand. This has been particularly true in cases where they found current on the pipes, as they feared the current might be harmful. From time to time there have been complaints of electric shock from water pipes on which electrical circuits have been grounded. More recently, a question has arisen as to whether such grounding might not be promoting certain corrosion troubles on pipes and fittings. In some cases, water companies came to suspect that somehow grounding might be responsible for objectionable tastes, odors, and discolorations of the water, instances of which they sometimes experience. The water companies may have been unduly apprehensive of harmful effects from grounding and disposed to attribute to it troubles for which, perhaps, grounding was not to blame. On the other hand, it seemed reasonable to have the facts investigated so that the effect of this practice of grounding on water pipes, whether harmful or not, might be better understood.

Accordingly, the American Water Works Association and the Edison Electric Institute jointly invited representatives of other interested groups to consider a cooperative study of this problem. As a result, the American Research Committee on Grounding was formed. Some fourteen national organizations are represented. The committee's objectives are set forth in a resolution adopted at the organization meeting, which reads as follows:

To investigate and report upon, as conclusively as may be: (a) electrical grounding connections, made to water, gas, or drainage pipes, with respect to the effects of any kinds of electrical current thereby caused to flow, upon (1) the pipes, (2) the contents of the pipes, and (3) fire and personal hazards; (b) conditions in the electrical circuits involved which produce current flow over pipes; (c) measurement of stray currents; (d) efficiency of electrical grounding.

Since that meeting, the committee has been making field and laboratory investigations. The results to date may be briefly summarized in the following paragraph taken from a progress report dated June 14, 1938:

To date the technical subcommittee has investigated some 26 field cases where complaints have been made and where it was suspected that stray alternating or direct current might have been a factor in producing the trouble. In none of the cases so far investigated, however, has the trouble been found to be due to alternating current on the water piping system, although on the other hand, the investigations have not shown that such current may not have contributed to the cause of the complaint in some of the cases.

A number of significant factors have been found, including the tendency of the water in question to corrode the piping or other metal, the effect of dissimilar metals in the piping system creating galvanic couples, and corrosion of hot water supply boilers due to abnormally high temperatures. Further field investigations are to be made, as cases are reported to the committee by water companies, where troubles such as corrosion, impairment of water, sparking, or electric shock, are suspected of being caused by electrical grounding. Field work is being supplemented by laboratory investigations of the fundamental electrochemical principles involved.

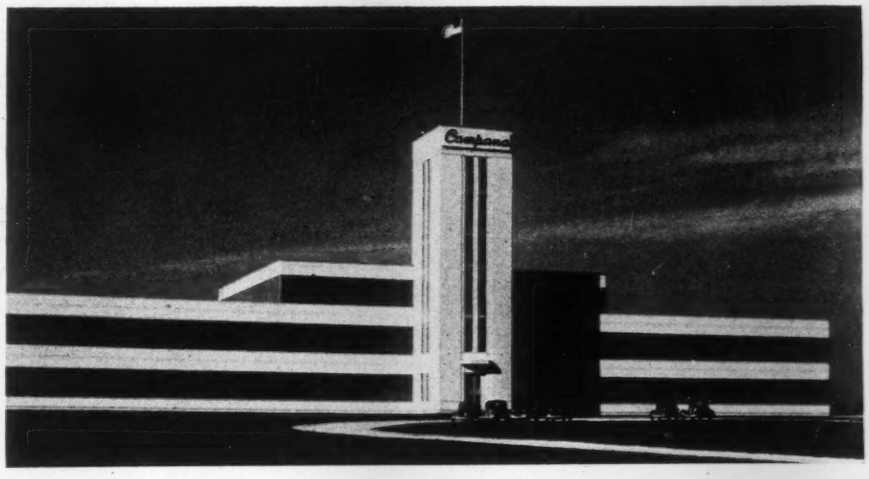
From this statement it appears that the troubles experienced in the field by the water companies have been due largely to causes other than the grounding of electrical circuits on their pipes, although, of course, this is not the case where electric shocks from water pipes have been experienced. However, the committee has had no case of electric shock called to its attention for investigation.

The committee is desirous of investigating as many cases as possible where troubles such as corrosion, impairment of water, or electric shock, are suspected of being caused by electrical grounding, and invites information about such cases. In order to save time and traveling expenses, investigations are preferably limited to the area within 100 miles of New York City, but reports of cases elsewhere will be welcomed. Word concerning cases should be sent to the committee's secretary, Charles F. Meyerherm, 511 Fifth Ave., New York City. Arrangements can then be made by which all parties interested can take part in any inspection or investigation which the committee may make.

*Consulting Engineer. Chairman, American Research Committee on Grounding.

Hot Water Heating System for New Campana Plant Has Novel Features

Well Water Provides Cooling and Dehumidification in Summer



By Samuel R. Lewis*

Year 'round air conditioning was specified for Campana's modern new building as an aid to efficient production and for employee comfort. Cooling and dehumidification of the air in summer are accomplished by means of cold well water; in winter, hot water is circulated through the same piping and convectors. . . . The water is heated by means of twelve oil fired boilers controlled in three groups of four each, cutting in and out automatically as required by the outgoing water temperature. . . . Mr. Lewis, consulting engineer for this unusual plant, describes the heating and air conditioning

THE Campana Sales Co.'s new plant at Batavia, Ill., incorporates almost every conceivable modern aid to efficient production of its product—various proprietary toilet articles. Situated on an extensively lanscaped site, the building is a windowless structure of reinforced concrete construction with exterior walls of terra cotta. Translucent bands of hollow glass blocks extend around the entire exterior.

Electric energy is the only public utility service used. All water comes from deep wells on the property, and sewage is treated in a private plant.

Year 'round air conditioning was specified, for the company's founder—Mr. E. M. Oswalt—believes in comfort for his employees as well as close control of temperature, relative humidity, and dust in the manufacturing processes. The air conditioning and the unique heating plant—a forced circulation hot water system with 12 automatic oil fired steam boilers for heating the water—are among the most interesting features of the project. The cooling and dehumidification are accomplished by cold water from deep wells; the same water circulating piping and convector system is employed the year around to deliver heat to the rooms in winter and to remove heat from them in summer. All of this piping is sweated copper.

Every department on the second and third stories has year 'round air conditioning from a plant arranged to filter and heat or cool the air, varying in proportion of outdoor to recirculation according to occupancy and usage. The volume of outdoor air taken in varies from a minimum of perhaps 10 per cent with 90 per cent recirculated in extremes of warm or cold outdoor weather to 100 per cent from outdoors during intermediate seasons when neither heating or cooling are necessary.

The first story is used exclusively for storage and shipping, and while it is heated in winter it is not cooled in summer. Certain manufacturing processes involve unusually low relative humidities and temperatures in some departments while others are those normal to any intensively occupied office building. All of these conditions have been met successfully by adjusting the rate of water flow and volume of air movement to the requirements of each department.

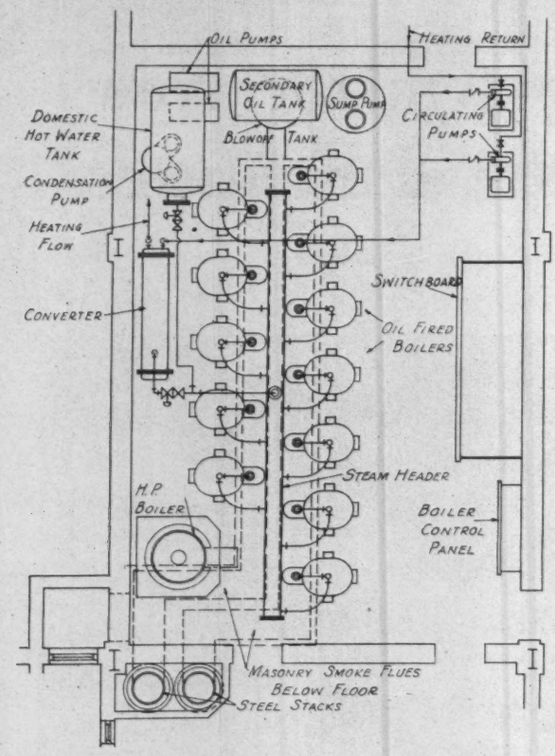
Heat is transmitted to toilet rooms and many small auxiliary rooms not requiring conditioned air by direct hot water radiators of the pressed steel high pressure type, wall hung.

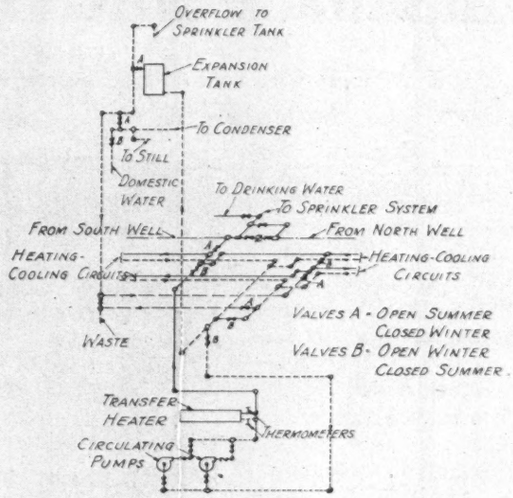
The Conditioning Units

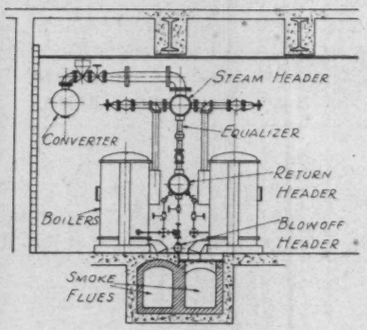
Each principal room on the second and third stories has one or more overhead horizontal conditioning units—which are of original design in that the exhaust fans as well as the supply fans are driven by a single electric motor. Each of these units is served by an outside air intake duct coming from a copper gooseneck hood on the roof and each has an outboard exhaust duct leading to the roof, and discharging close to and above the intake. No contamination of the entering air due to this relation has been observed.

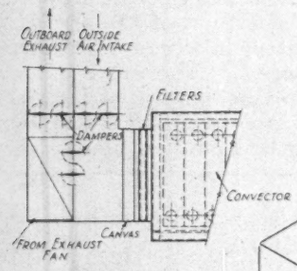
The supply and exhaust systems are connected through interlocked dampers in such manner that the exhaust fans may deliver any adjusted proportion of the air back through the filters and convectors for recirculation or may discharge it directly outdoors.

*Consulting Engineer. Member of Board of Consulting and Contributing Editors.

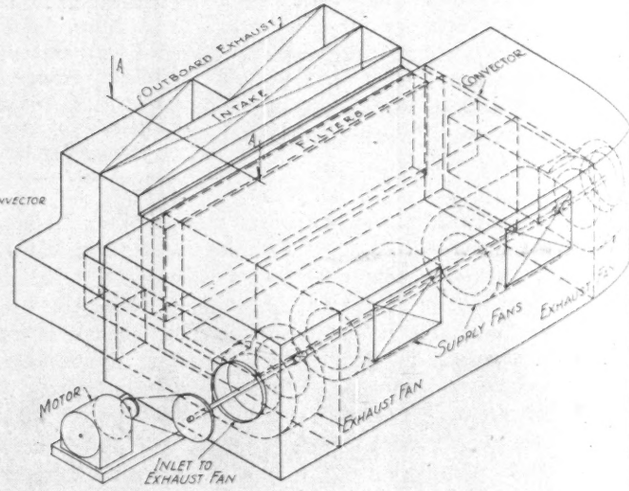

OIL PUMPS
HEATING RETURN
SECONDARY OIL TANK
SUMP PUMP
DOMESTIC HOT WATER TANK
BLOWOFF TANK
CIRCULATING PUMPS
CONDENSATION PUMP
HEATING FLOW
SWITCHBOARD
CONVERTER
OIL FIRED BOILERS
STEAM HEADER
H.P. BOILER
BOILER CONTROL PANEL
MASONRY SMOKE FLUES BELOW FLOOR
STEEL STACKS

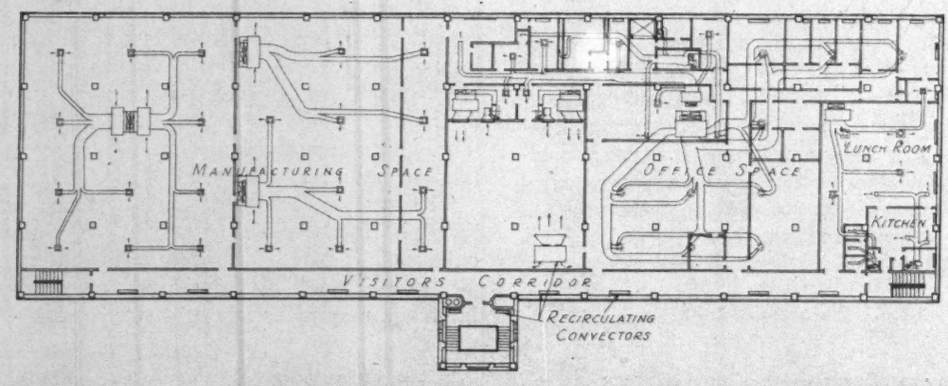

OVERFLOW TO SPRINKLER TANK
EXPANSION TANK
TO CONDENSER
TO STILL
DOMESTIC WATER
TO DRINKING WATER
TO SPRINKLER SYSTEM
FROM SOUTH WELL
FROM NORTH WELL
HEATING-COOLING CIRCUITS
HEATING-COOLING CIRCUITS
VALVES A - OPEN SUMMER CLOSED WINTER
VALVES B - OPEN WINTER CLOSED SUMMER
WASTE
TRANSFER HEATER
THERMOMETERS
CIRCULATING PUMPS


STEAM HEADER
EQUALIZER
CONVERTER
RETURN HEADER
BOILERS
BLOWOFF HEADER
SMOKE FLUES


OUTBOARD EXHAUST
OUTSIDE AIR INTAKE
FILTERS
DAMPERS
CONVECTOR
CANVAS
FROM EXHAUST FAN

SECTION A-A


A
OUTBOARD EXHAUST
INTAKE
FILTERS
CONVECTOR
SUPPLY FANS
EXHAUST
MOTOR
EXHAUST FAN
INLET TO EXHAUST FAN


MANUFACTURING SPACE
OFFICE SPACE
LUNCH ROOM
KITCHEN
VISITORS CORRIDOR
RECIRCULATING CONVECTORS

Opposite page, upper left—Plan of the boiler room . . . . . *Upper right*—Diagram of the principal water piping . . . . . *Left center*—Cross section through boiler room . . . . . *Right center*—Typical supply and exhaust fans, with convectors and ducts . . . . . *Bottom*—Plan of second story

In general the air conditioning units are situated in the rooms which they serve, partly concealed in the furred-down ceilings with their bottoms and bearings always accessible. The air distributing ducts are concealed above a suspended ceiling; the spent air returns without ducts directly to the exhaust fans. The temperature is controlled by electric thermostats which vary the volume of water flowing through the convectors in accordance with the individual requirements, and which are readjusted and reversed, as the seasons change, by an outdoor master thermostat.

Every air conditioning unit has a water-tight, dew-collecting tank draining through copper tubing to a visible outlet above a trapped catch basin in the floor. Many rooms, such as corridors, have floor mounted convector units equipped with noiseless recirculating fans, all of which are controlled automatically and all of which have dew drains, since they receive cold water in summer as well as warm water in winter.

The Basement Equipment

All piping, conduits, drains, etc., are overhead near the first story ceiling, and there are no trenches or buried pipes in the building except certain sewers which serve the first story. There is no basement beyond a zone extending across the building at its center, approximately the same width as the tower. This basement holds the boilers, water circulating pumps for use in winter, and the electrical distribution board, as well as large tanks storing various liquids used in the manufacturing processes. Fuel oil is stored in two 7500 gal steel tanks outside of the building and alongside the railway and trucking areas. There is one high pressure, oil fired steam boiler used for process purposes and entirely separate from the heating system. The 12 low pressure heating boilers are arranged in two batteries, back to back, with their breechings directly connected into two underfloor masonry ducts which lead to duplicate steel stacks in one corner of the tower.

The boilers are controlled in three groups of four, each cutting in and out automatically as required in response to the outgoing water temperature demand. They have electric ignition, with the usual time delay and thermal protective devices. A very complete installation of recording and indicating instruments has been made.

Since the deep well water supply was likely to carry a considerable volume of salts precipitable when heated above a rather low critical temperature, the 12 boilers all operate on a closed steam system, with extra large steam and return headers. The low pressure steam never leaves the boiler room, and the condensate is all returned to the boilers by duplicate automatic pumps. There are also duplicate automatic oil pumps for delivering the fuel oil from the outdoor storage tanks to a small auxiliary oil tank in the boiler room.

The steam is used to warm the recirculated water of the heating system in a copper tube transfer heater arranged for easy cleaning. The domestic hot water for process and for the large plant cafeteria also is heated by steam. Since the temperature of the recirculated heating system water responds directly to the heat demand, it is possible to control the number of boilers in operation by a master step-aquastat in the departing water.

The process boiler has both steam and electric feed pumps and serves certain small process requirements, including a water distillation system. The make-up water for this boiler comes from a conventional zeolite softener, and is mixed judiciously with enough raw water to protect the boiler against pitting.

The steam piping inside the boiler room mostly is of screwed and welded steel, but all water circulating piping and much of the process piping is of sweated hard copper.

The Well Water System

All water comes from two drilled wells, one at each end of the building and about 325 ft apart, one (No. 1) 500 ft deep and the other (No. 2) about 1000 ft deep. No. 1 delivers water at 52.5 F, while the water from No. 2 is about 1 F warmer. Each has a vertical turbine pump controlled automatically to maintain a standard pressure within a 6 ft diameter by 20 ft long underground steel storage tank. The tanks are built for 150 lb pressure and are well insulated in thoroughly drained soil far below the sun heated surface. They have the usual pressure gages and relief valves, all accessible at the tank ends which project into the well pits.

The two pneumatic water tanks with their deep well pumps supply all water for domestic use and for fire protection as well as for air conditioning in summer. They are connected by a 5 in. header running the full length of the building with an automatic pressure actuated valve in the header which normally is closed, keeping tank No. 2, which is filled from the deeper well, normally at the full pressure of 60 lb ready to serve, while the other tank and the shallower well with its cooler water carries the load. Should the pressure in the general house system fall to 40 lb the automatic valve opens and well No. 2 goes into action to supplement well No. 1, stopping only when the pressure is again built up.

There is a manually operated bypass valve around the automatic valve for emergency use.

The 40,000 gal sprinkler tank in the tower is served by a 6 in. riser from the main header on the side of the above valves toward well No. 2 with the usual float control to maintain a constant water level in the tank. In case of fire, two 5 in. connections with automatic pressure actuated valves connect directly from the respective wells to the sprinkler system, bypassing the sprinkler tank and putting both well systems into parallel operation.

When the cold well water is used for air conditioning the water (except that for a small line serving drinking fountains) goes first to the cooling convectors of the second and third stories, from them to the convectors of the first story, and then serves the domestic and sprinkler tank requirements before overflowing at the top of the latter for further use for lawn sprinkling before being wasted to the storm sewer.

Most of these functions are automatic, with interlocked electrically operated valves, since in winter all

Interior view showing one of the overhead units

communication between the pumped hot water heating system and the domestic system is terminated. In winter the heating system water is in a closed circuit, with a steel pneumatic expansion tank in the tower, and is sent through the steam heated transfer apparatus in the boiler room by duplicate centrifugal pumps. The circulated water in winter is protected against possible freezing by addition of antifreeze compound since, when large volumes of air directly from outside are introduced, there is possibility of freezing. The antifreeze solution is stored from year to year. The expansion tank being closed, it is possible to heat the water to 250 F or warmer, without boiling, should this be necessary.

The temperature of the cooling water in summer automatically is controlled by regulating the volume of well water passing the convectors through a humidistat. In winter the water temperature is varied automatically in proportion to the weather through an outdoor thermostat.

The system is designed to permit and to maintain exact year around temperature and relative humidity and dust control in each of the many separate divisions; the modified unit system of air conditioning gives this flexible, separate control admirably.

All chemical hoods, special process rooms, the toilet rooms, and the cafeteria kitchen have independent exhaust fans. Certain comparatively small storage rooms for raw material have independent automatically controlled mechanical refrigeration plants, due to need for refrigeration the year around during the seasons when no general refrigeration is necessary.

Certain processing rooms are equipped throughout with explosion proof electrical appliances, though no explosions have ever occurred in the former unventilated, conventionally heated plants of the company.

Among the lessons gained by experience with this installation are that the water heating-cooling convectors should be subdivided by valves and arranged so that about half of the surface can be cut out and drained in winter.

Long lines of hard copper tubing must have unusual arrangements to control expansion and contraction, preferably by loops.

Intake and outlet dampers should be very tight, preferably closing against rubber hose attached to the frames, since with a subzero wind blowing toward an intake the shutdown temperature to leeward of closed, conventionally built dampers drops rapidly.

All dampers should be spring operated to shut off outside communication when the respective fans are shut down, and all valves which control fluids which could freeze in case of failure of the automatic controls should be spring opened in case of failure of these controls.

Architects for this project were the late Frank D. Chase, and Childs & Smith. The contractor for the heating and cooling system was Mehring & Hanson Co. The author was consulting engineer.

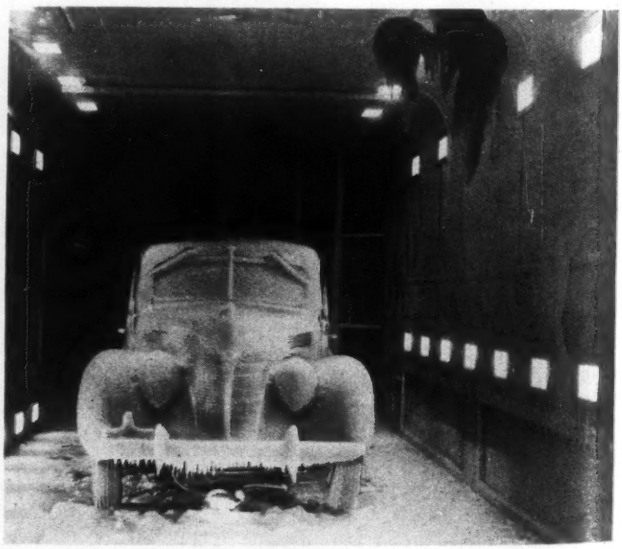
Huge Weather Tunnel Tests Fords

A huge new "weather" tunnel, said to be the first of its kind ever built solely for scientific research in motor car design, has been put in operation at the Ford Motor Co. engineering laboratory, Dearborn, Mich. Purpose of the tunnel is to make possible reproducing in a laboratory any kind of weather to be found anywhere on earth any time of year.

The 500 ton refrigeration system can maintain a temperature of 20 deg below zero in the tunnel. There is also a cold chamber in which temperatures as low as 40 below zero can be maintained for tests of engines, oil, batteries and starters. The temperature in the tunnel can be varied from 20 deg below to at least 150 deg above zero, atmospheric pressure can be adjusted to duplicate any climatic condition, and the moisture content and oxygen content of the tunnel air can be controlled. The air is cooled by refrigerating coils built inside the tunnel.

The tunnel itself is a rectangular structure 124 ft long by 35 ft wide and 14 ft high, enclosing an elliptical chamber 10 ft in diameter. In this chamber air can be circulated at velocities as high as 85 mph, the air being driven by a 400 hp electric motor turning a 10 ft three blade propeller.

A Ford V-8 sheathed with ice under test in the tunnel



The Theory of Evaporative Cooling

By W. S. Bodinus*

Because of the number of questions pertaining to evaporative cooling received by the editors from readers, we asked Mr. Bodinus to prepare this elementary discussion of the principle involved. . . . Those unfamiliar with this type of air conditioning will find his article a clear explanation of its basis

EVAPORATIVE cooling represents one of the oldest practical applications of air conditioning principles. Systems of the water spray air washer type for air cleaning and humidification were used even before the 20th century, and it was observed that a considerable cooling effect could be obtained under certain outside temperature and humidity conditions. Historical accounts of the ancient Egyptians, the Indians, the Chinese and others illustrate methods of evaporative cooling by sprinkling water on reed carpets that closed entry ways to living chambers. Natural air currents passing over the wetted surface caused evaporation of water and cooling of the air.

Adiabatic Saturation

One of the air conditioning principles that seems most difficult to understand is the theory of adiabatic[1] saturation of air, first rationally expressed by W. H. Carrier. This theory is based on the conversion of sensible heat of air to latent heat while the total heat remains constant. If a continuous quantity of air at a high dry bulb temperature and a low dew point temperature is passed through a dense spray chamber of recirculated water, the water will take a temperature the same as the wet bulb of the entering air; the condition of the air leaving the spray chamber will be saturated (100 per cent relative humidity) at the spray water temperature (which as stated before is the same as the entering wet bulb temperature). The water remains at a constant temperature, although a small portion of it is evaporated to add the necessary amount of latent heat to the air. The heat required for the evaporation of this small amount of water is removed from the air by cooling the air from the entering dry bulb to the leaving saturated condition.

Thus the air which entered the spray chamber leaves the chamber having had its dry bulb temperature reduced and its dew point increased. In this process there has been no extraction or addition of heat (adiabatic). Note that the process is dependent only upon the wet bulb temperature of the entering air, and both the water and the leaving air finally take a temperature the same as the entering wet bulb. Naturally the thoroughness of this process depends upon the efficiency of the spray chamber, which in turn is governed by the density of the spray area, atomizing efficiency of the nozzles, velocity of the air through the spray, and the evenness of the distribution of the air through the chamber.

Influence of Outside Conditions

Many times in spring and fall, and once in awhile during summer, conditions of the outside air are such as to allow the use of the evaporative principle for comfort cooling provided the proper temperature control is used to compensate for the varying outside conditions. For evaporative cooling purposes a simple control is a thermostat situated in the path of the air leaving the spray chamber which actuates automatic outside air dampers and automatic return air dampers. This thermostat is known as a "dew point thermostat" although it actually causes a mixture of air entering the spray chamber to be at the correct entering wet bulb so that the phenomenon of adiabatic saturation can take place.

If it is desired to cool a conditioned space to an average condition of 80 F dry bulb and 50 per cent relative humidity, the room dew point must be 59½ F. Since there is always a certain latent heat pick up, the dew point of the air entering the room must be lower than 59½ to compensate for this—so the air is introduced into the room at a dew point 1 to 10 F lower than actual room dew point. For the ordinary office space, the entering dew point might average 56 F.

If the principle of evaporative cooling is to be used in this case, naturally the air leaving the spray chamber must be at a temperature no higher than 56 F. In order to do this, one of the combinations of air entering the spray chamber listed in Table 1 would be necessary.

Table 1—Entering Air Conditions to Spray Chamber for Room Condition of 80 F and 50 Per Cent

Dry Bulb	Dew Point	Relative Humidity	Wet Bulb
90°	16°	6%	56°
85°	26°	11%	56°
80°	34°	19%	56°
75°	40°	29%	56°
70°	45°	40%	56°
65°	49°	60%	56°
60°	53°	78%	56°
56°	56°	100%	56°

Obviously, from Table 1, atmospheric conditions do not exist in midsummer in most areas to satisfy the necessity of a 56 F dew point leaving the apparatus, although the values from 75 F dry bulb down to a 56 F saturated condition do exist quite often in spring and fall. For installations where the internal heat loads (lights, people, motors) are great (and these are often coupled with sun effect) there is a necessity for cooling during intermediate seasons or even in winter which can be very well satisfied by the evaporative cooling system.

The design condition of 80 F and 50 per cent relative humidity for satisfactory room comfort conditions is ordinarily based on a very mild movement of air, say a velocity of 15 to 25 fpm in the occupied space. By increasing

*Engineer, Carrier Corp.

[1]Pertaining to or designating processes not accompanied by gain or loss of heat. In the adiabatic saturation of air, the total heat remains constant; some of the sensible heat of the air is converted to latent heat, which lowers the dry bulb temperature of the air and increases its dew point temperature and of course its relative humidity.

Evaporative cooling spray chambers and fans for bakery cooling

the movement of air in a given space by circulating more air (increasing the velocity in the space to, say, 100 fpm) a greater cooling power is obtained. The same net comfort cooling result at the increased air movement can now be obtained with a design condition of 80 F and 65 per cent relative humidity, although the atmosphere would have a somewhat damp and musty feeling. It is possible to increase the cooling power of the air even further by increasing the velocity of the air circulation in the room, but objections because of excessive air movement and drafts usually limit the velocity in the space to less than 100 fpm. Then, too, relative humidities much higher than 65 per cent cause a very damp atmosphere which is objectionable from a comfort standpoint as well as possibly being detrimental to products on display and injurious to furniture, fixtures, decorations, etc.

It cannot be stressed too strongly, in my opinion, that before conditions using 65 per cent relative humidity combined with room velocities of around 100 fpm can be considered acceptable by the air conditioning industry and the public, extensive field tests will be needed. Such conditions have proved acceptable in fairly complete laboratory tests, but field installations have been too few and restricted to permit any conclusions to be drawn to date.

The room condition of 80 F dry bulb, 65 per cent relative humidity, and 67 F dew point could be satisfied if the dew point of the air leaving the spray chamber were approximately 65 F (allowing 2 F lower dew point of the air entering the room than room dew point). Therefore the wet bulb of the mixture of air entering the spray chamber must be 65 F. In order to do this the air conditions entering the spray chamber of Table 2 would be necessary.

Table 2—Entering Air Conditions to Spray Chamber for Room Condition of 80 F and 65 Per Cent

Dry Bulb	Dew Point	Relative Humidity	Wet Bulb
100°	39½°	12½%	65°
95°	45°	18%	65°
90°	49°	24%	65°
85°	53°	33%	65°
80°	56°	45%	65°
75°	59°	58%	65°
70°	62°	78%	65°
65°	65°	100%	65°

From Table 2 it again can be seen that atmospheric conditions do not usually exist in mid-summer to satisfy the necessity of a 65 F dew point leaving the apparatus. The field of usefulness of the evaporative cooling principle has however been broadened (on the assumption of a higher room relative humidity) in that a greater number of days will fall within the values of Table 2. There are a few places in some of the western states where atmospheric conditions exist to make the evaporative cooling system satisfactory during the major part of the summer period. An excellent system which has been used satisfactorily in many of these areas is a combination of refrigeration and evaporative cooling.

Good engineering practice in designing a complete air conditioning system requires a thorough consideration of the variance of outside weather conditions. It is not economically sound to design the cooling equipment for a maximum observed condition that may occur only once or twice a year and most installations are designed for an average summer condition which is somewhat less than the peak condition. Even with this original discount, the use of refrigerating equipment, or some means of latent heat removal, is required throughout the summer season in most parts of the United States. The use of spray type air washer conditioning equipment which will make use of the principle of evaporative cooling does find a very economical and practical application for intermediate use, as well as affording complete humidification for winter operation.

Air Conditioning a Production Factor

The air conditioning installation in the plant of the Warren Telechron Co., Ashland, Mass., where thousands of small clock motors are produced weekly, offers an example of benefits which accrue to workers and product. Dust removal, humidity control, and adequate ventilation were prime factors to be considered in the design of the 9600 cfm system installed in the assembly section by Lawton Engineering Corp., of Boston.

During the fall season thousands of rotor units are produced in the air conditioned room. The greater volume of the Telechron business is produced on motor conveyors, and special motors for heavier instruments and larger clocks are also assembled in the room on separate benches. Failure to eliminate dust, dirt, and foreign grease from the interiors of the sealed-in motors might result in excess wear and motor failure. Inspection and assembly work is therefore close and difficult, and favorable working conditions are a prime consideration.

The system consists of a 25.6 ton compressor driven by a 20 hp motor connected to an air conditioner, which provides conditioned air distributed from ceiling ducts. A mean relative humidity of 50 per cent is maintained.

An interesting angle of the installation involves a testing rack where 8000 Telechron motors are suspended. Each consumes only two watts of electricity, but this factor increased the heat gain by approximately 4⅔ tons.

Noise Measurement

By H. F. Hagen*

A Clear and Concise Explanation of the Principles of Noise Measurement

ANYONE possessing a knowledge of the fundamental laws of physics, which form the basis of all engineering, can readily extend this knowledge to include an understanding of the principles of noise measurement. He may be troubled at first, however, in relating this new matter to his previous knowledge by the seemingly new quantities that appear in the formulas and the unfamiliar groupings that even old, familiar units can develop.

Our most important equation, the one for sound intensity in ergs per second per square centimeter, introduces both new quantities and peculiar groupings.

$$I = \frac{P^2}{\rho C} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where

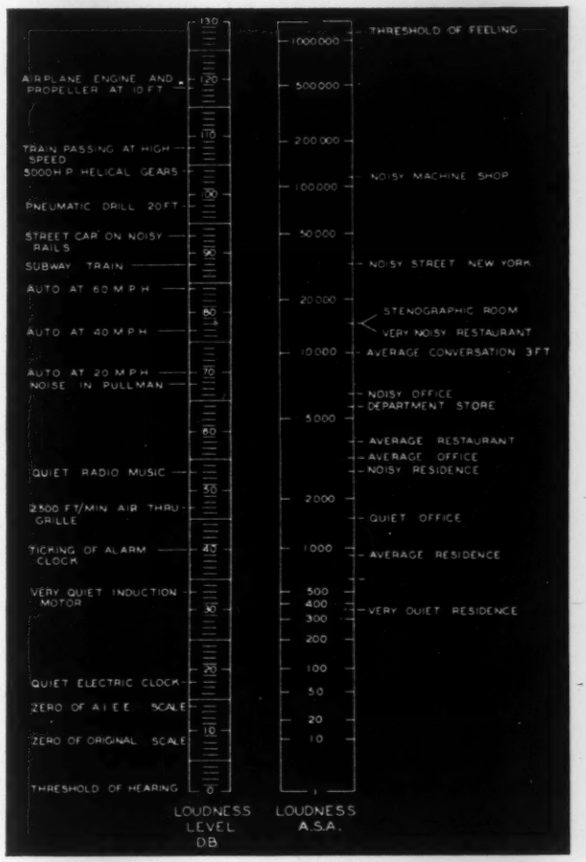
P is the sound pressure in dynes per sq cm;
ρ is air density in grams per cc;
C is the velocity of sound in cm per sec.

A complete discussion of this equation and its terms will show that there is really nothing new involved. The equation depends entirely upon combinations of the usual units of force, mass, length, and time.

The use of the centimeter, gram, second (CGS) units has been definitely established in sound literature, and brief definitions of these units may be welcome to those who may have forgotten them. Force, mass, length and time are the primary units in any system. In the CGS system the unit of time is the familiar second, the unit of length is the centimeter, the unit of mass is the gram, and the unit for force is the dyne, a force which acting on a 1 gram mass will produce an acceleration of 1 cm per sec per sec.

Other units of measurement are derived from these four primary units, such as:—Pressure in dynes per square centimeter, or bars. The latter term is frequently used in sound engineering. Work or energy in ergs. An erg is the work done by a force of one dyne moving through a distance of one centimeter.

In Equation 1 the intensity I is measured in ergs per second per square centimeter. The intensity of a sound is, therefore, the quantity of sound energy per second passing through a unit area. For the interest of those who like a little mathematics a readily understandable derivation of Equation 1 is given in "Acoustical Engineering" by W. West, published by Sir Isaac Pitman & Sons Co. At present it is sufficient to say that the formula results simply enough from the application of acceleration, force, and work to a wave motion.



Loudness level and loudness scales

Modern acoustical engineering has been developed with scientific precision. The telephone industry has led the way under the direction of Dr. Harvey Fletcher of the Bell Telephone Laboratories. As might be expected from such origin, emphasis has been placed on the effect of sound on the human ear, and the measurements of sound are designed to approximate ear reaction and only indirectly to give physical quantities.

In general the response of any of the human senses to an external stimulus is in terms of a ratio of the added stimulus to the previously existing conditions or level affecting that sense. At night, for example, we have a low light level and a single 50 watt bulb gives us a lighted room. But in the same room in the daytime, when we light the bulb because of dark clouds outside, the lamp seems to help only slightly. In this case the added stimulus, the light of the lamp, produced a smaller sensation of change.

The ear and sound show a similar relation. When the noise level is low, as in a quiet room, a small amount of noise, such as is made by a desk fan, is not only noticeable but unpleasant. In a moving train, however, with a high noise level the noise of the same fan is barely audible.

This peculiar physiological response to stimuli obeys very closely the result of integrating the mathematical

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expression of the change divided by the level and gives: Sensation=log (I_2/I_1). That is, the sensation is equal to the logarithm of the ratio of the intensities of the stimuli.

In acoustics the unit log of ratio is called a *bel*, and one tenth of this unit is the *decibel* (db). We then have: bels=log (I_2/I_1), where I_2 and I_1 are sound intensities. The common logarithm, base 10, has been adopted. By definition the decibels are 10 times as many, so:—

$$\text{db} = 10 \log \frac{I_2}{I_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [2]$$

The decibel happens to represent the smallest change the ear can perceive. Decibels may also be expressed in terms of sound pressure ratios. From Equation 1 substituting for I in Equation 2 we have—

$$\text{db} = 10 \log \frac{\frac{P_2^2}{\rho C}}{\frac{P_1^2}{\rho C}}$$

and cancelling the ρC

$$\text{db} = 10 \log \frac{P_2^2}{P_1^2}$$

and bringing the square into the coefficient

$$\text{db} = 20 \log \frac{P_2}{P_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [3]$$

You are apt to meet in your work decibels based on either intensities or pressures and you might be puzzled by the coefficient 10 for intensities and 20 for pressures. Both the intensities and pressures in the equations are the square roots of the average of the squares of the instantaneous values of their actual wave quantities and are usually referred to as root mean square, or RMS, values.

Sound Meters

Accurate meters have been developed that measure the ratio of sound pressures in decibels. These machines consist of a microphone, preferably of the condenser type, amplification of the microphone impulses, and a meter to measure the electrical energy. Earlier instruments depended on the matching of two sound intensities, one of known magnitude, by human hearing; but this method has been almost completely discarded.

In using the modern instrument, the acoustimeter, to measure the noise of a fan, motor or other apparatus, a reading is first taken of the room level with the fan not running. Then another reading is secured with the fan in operation. It is apparent that, if I_1 is the actual intensity of the room noise, our first reading must be 10 times the common logarithm of the ratio of this I_1 to some reference intensity. I_0. This I_0 is the reference level the meter is calibrated and designed to work from. Unfortunately there are at the present time three reference levels in use by instrument manufacturers.

One is based on a sound pressure of one millibar, or 0.001 dynes per sq cm, chosen arbitrarily for the convenience of the round figures. The second is based on the minimum audible intensity of sound, called the threshold of audibility; but also, unfortunately, this quantity has varied in different experiments, and some instrument makers place it at a sound pressure of 0.425 millibars and some at 0.6 millibars. (The millibar is one thousandth of a dyne per sq cm). The American Standards Association has suggested a reference level of 0.202 millibars—again for the convenience of a round number, 1×10^{-16} watts per sq cm, which appears when intensities instead of pressures are used.

These references vary by approximately seven decibel intervals, so that a sound of 30 decibels on the A.S.A. reference of 1.0×10^{-16} watts (decimal point fifteen zeros and then 1) would be 23 db from the threshold of audibility reference and 16 with the one millibar instrument. This same difference of 7 db between the three bases holds no matter what the amount of the decibels may be, so it is a simple matter to transfer from one to another.

Until this matter has been standardized, it is apparent that the decibels of our room sound means nothing unless we give our reference level, or "zero" as it is usually and somewhat carelessly termed.

[A second article will give examples of calculations and specifications.]

Study Air Conditioning Costs

RESULTS of a preliminary survey on the cost of air conditioning were reported to the recent annual convention of the National Association of Building Owners and Managers by the chairman of the joint committee on air conditioning, William E. Malm. The study was based on data obtained by questionnaires on 20 installations of 100 tons or over and 22 installations of less than 100 tons. It was stressed that definite conclusions should not be drawn from the figures presented because of the limited data available and relatively brief operating experience. The survey is to be continued.

For the plants of 100 tons and over it was found the average area served per ton of capacity was 300 sq ft; the average cost per ton of capacity was $448, the average cost of installation per sq ft was $1.50, and that the average size of these plants was 223 tons. Annual operating and owning cost was determined to be 29.8c per sq ft, including 18.1c per sq ft for capital requirements.

For the plants of less than 100 tons the average area served per ton of capacity was 380 sq ft, the average cost per ton of capacity was $428*, the average cost of installation per sq ft was $1.13, and the average size of these plants was 32.9 tons. Annual operating and owning cost for the installations of less than 100 tons was found to be 31.1c per sq ft, allowing 13.4c per sq ft for capital requirements.

Check Survey of 26 Proposals

A check study was made based on information furnished by a large manufacturer of air conditioning apparatus which represented proposals they had made for the installation of air conditioning apparatus in 26 buildings involving 1,607,000 sq ft of net rentable area. The average size of these installations was 153 tons, the average area served per ton of capacity was 403 sq ft, and the average cost of installation per ton of capacity was $364. In these 26 proposals the installation cost ranged from $278 to $703 per ton of plant capacity.

*It is not known why the cost per ton of the smaller plants covered in the study was less than the figure for the larger ones.—Ed.

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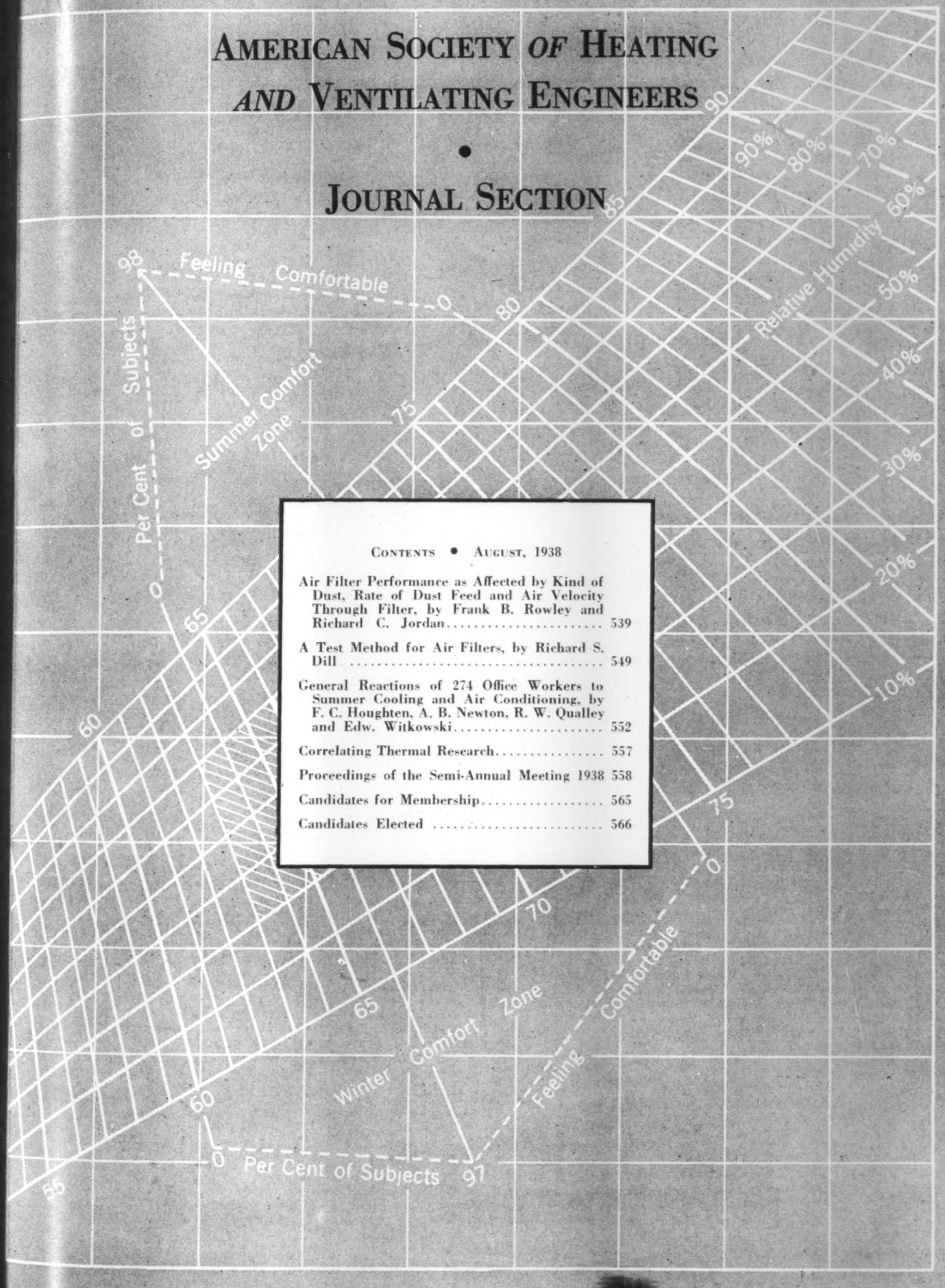

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Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity Through Filter

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Engineering Experiment Station, University of Minnesota.

IT is difficult to set up a laboratory test procedure which will measure the characteristics of an air filter and give reliable data by which its performance on a particular type of work can be predicted. This is due partly to the lack of a simple, standard method of measuring the amount and quality of dust in the air, and partly to the difficulty of selecting test conditions which will simulate practical requirements for a filter.

In 1933 the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS adopted a Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. This code provides a method of performing an accelerated service test on a filter. A dust mixture prepared to represent the average dust found in air is mixed into the air stream leading to the filter. The amount of this dust in the air is measured before and after passing through the filter. Air velocity and limiting pressure drop across the filter are selected to meet average filter requirements.

In formulating this code there were many variables to consider and part of the conditions had to be selected arbitrarily to make up a workable test procedure. Some of the controversial questions were:

1. A suitable method of measuring the dust concentration in the air entering and leaving the filter.
2. The preparation of a synthetic dust mixture which would have the properties of a dust found in average air.
3. The maximum allowable dust concentration that could be used for the air entering the filter without handicapping the performance of the filter.
4. A representative air velocity through the filter during the test period.

Methods of Measuring Dust Concentration in Air

An accurate method of measuring the amount of dust in the air before and after passing the filter is an essential requirement of a test code. Several methods have been proposed, but no single method gives all of the properties of the dust which should be known for a complete analysis. Of these various proposals, the weight method was finally selected as the best adapted to an accelerated laboratory test. In using this method a known weight of dust is mixed with the air entering the filter, and the weight in the air stream leaving the filter is measured by passing a small percentage of this air stream through a fine grain alundum crucible. With careful laboratory technique this method gives accurate results, but it does not give any information as to dust particle size nor is it a practical method to be used outside of the laboratory or for ordinary dust concentrations in the air. It is a decided handicap to confine all filter tests to tedious laboratory procedure, and it would be very desirable if some other method could be devised for measuring dust concentration which would be practical as a field test and which might be substituted for, or else be used parallel with, the present code method.

Synthetic Dust

The standard synthetic dust selected for the A.S.H.V.E. Air Filter Code consists of 50 per cent by weight of powdered lampblack containing a minimum of 97.5 per cent free carbon and having a minimum bulk value of 3.5 lb per cubic foot, mixed with 50 per cent by weight of Pocahontas bituminous coal ash screened to pass a 200 mesh screen. There are many varieties of dust in the air. The density, particle shape and size, as well as other characteristics, vary through wide ranges. Experience has shown that the type of dust passed through a filter may have a marked effect on its arrestance, and that a given dust may not have a like effect on the performance of two different types of filters. It is generally agreed that the dust in average air contains far less than 50 per cent by weight of carbon black. From this it is evident that the synthetic dust mixture as specified by the Code should have further investigation, and should be modified to simulate more nearly the dust which must be handled by the average filter.

Dust Concentration in the Air

The Code specifies a dust concentration of 0.35 g (gram) of the mixture per 1,000 cubic feet of air pass-

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ing through the filter. This represents a much greater weight of dust per unit volume of air than would be found under ordinary conditions, but it is necessary to increase the weight in order to shorten the time of a filter test, and also to have a dust concentration in the air leaving the filters which can be cleaned out of the air and weighed by practical methods. The criticism of this part of the Code has been due to the uncertainty as to the dust arrestance of some types of filters when using heavy concentrations of dust as compared with the arrestance of these filters under normal operating conditions in which very much lighter dust concentrations would be used.

It has been contended that when heavy dust concentrations were used the viscous fluid on certain types of filters would not penetrate the initial layers of dust rapidly enough to make them efficient in catching the succeeding dust particles, whereas under ordinary operating conditions more time would be available for the dust particles to be penetrated by the viscous fluid, and thus a continuously active dust catching surface would be provided. This is a question which can be settled only by more experimental work.

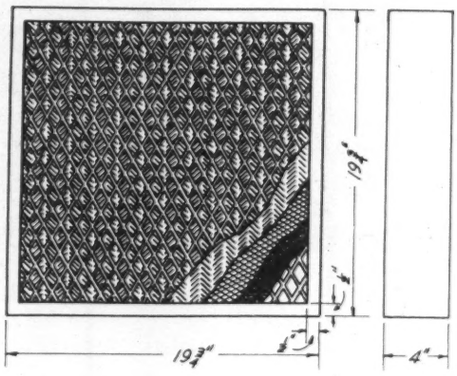
Air Velocity Through Filter

The optimum air velocity is probably different for different types of filters. For stationary filters the code specifies from 250 to 350 fpm through the gross face area of the filter, depending upon the type. There is a tendency in some types of apparatus to increase these velocities on account of the limited area available for installation of the filters. The increased velocity has a direct effect on the air pressure drop across the filter, and may have an effect on the arrestance of the filter. The effect on arrestance cannot, however, be predicted with the same certainty as the effect on air pressure drop.

Tests on Railroad Air Filters

In the spring of 1937 the Research Committee of the *Association of American Railroads* decided to make a series of tests on those filters which were in common use for air conditioning work in railroad cars. Preparatory to these tests representatives of all filter manufacturers interested in railroad air filter work were invited to a conference to discuss the advisability of making certain changes in the test procedure as outlined by the Code. As a result of this conference, together with the results from an examination made of several samples of dust collected from railway air conditioning systems, a new dust mixture was selected and other slight modifications were made in the test procedure.

The dust mixture selected consisted of 50 per cent by weight of Pocahontas ash screened through a 200 mesh screen, 20 per cent by weight of Illinois fly ash screened through a 200 mesh screen, 20 per cent by weight of lampblack meeting the Code specifications, and 10 per cent by weight of Fuller's earth. These percentages were mixed together and screened through a 100 mesh screen to insure uniformity of mixture. The dust mixture was fed into the entering air at the rate of 0.4 g per 1,000 cu ft of air passing through the filter, and the



Pack Arrangement	
Screen No.	Construction
1	Large Mesh Retaining Screen
2-5	Expanded Metal-Corrugated for Depth
6-11	Expanded Metal
12	Same as 6-11 but with three horizontal ribs ¾" in depth
13,15,17, 19,21,23	2 Layers copper mesh screen - 60 openings per sq.in.
14,16,18, 20,22	Same as 6-11
24	Rear Retaining Screen

Fig. 1—Construction details of filter *A*

rate through the filter was set at 300 fpm face velocity. Thus the greatest deviation from the Standard Test Code was in the dust mixture selected. Later a test was devised for rating a filter on its capacity to remove lint from the air. This test, however, has no relation to the present investigation.

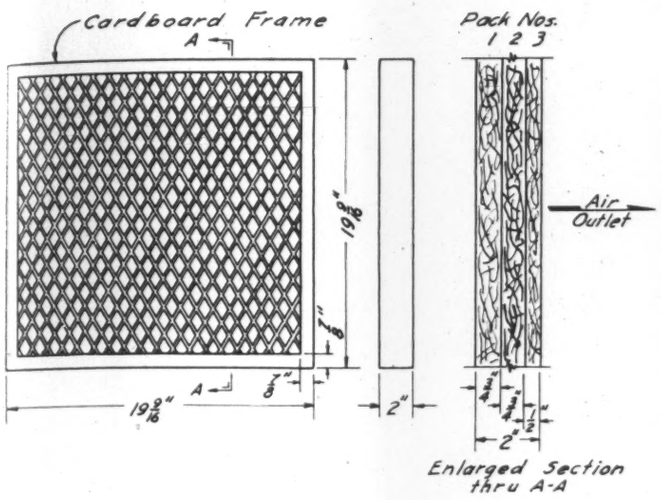
Extent of Present Investigation

The many questions which have arisen in regard to the adequacy of the Standard Test Code for determining the operating characteristics of different types of filters have shown the need for an extended investigation covering several phases of the subject. The present investigation covers three major points.

1. The relation between dust concentration in the air and filter arrestance.
2. The relation between air velocities through the filter and filter arrestance.
3. The relation of different types of dust and dust mixtures to filter arrestance.

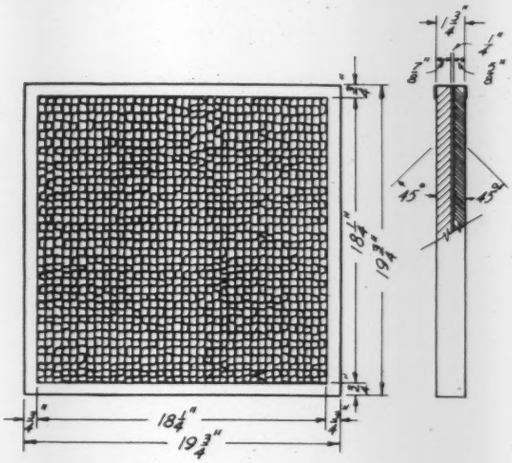
In making the investigation four commercial filters were selected as representative of four types in common use. These four types of filters were used throughout each series of tests and designated as Filters *A*, *B*, *C*, and *D*.

Filter *A*, shown in Fig. 1, is a recleanable type made up of 24 layers of metal screen as indicated in the sketch. These screens were graded in mesh and arranged to give a progressive pack for the filters. The filter was reconditioned between tests by thoroughly cleaning with trisodium phosphate, thoroughly drying, and dipping in a bath of filter oil at 150 F. The filter oil was allowed



Pack No.	Thickness	Fiber Diameter	Fiber Length	Condition
1	¾"	252 Microns	12.25 ft./cu.in.	Oiled
2	¾"	252 Microns	12.7 ft./cu.in.	Heavily Oiled
3	½"	135 Microns	14.2 ft./cu.in.	Oiled

Fig. 2—Construction details of filter *B*



Description	Pass No.1	Pass No.2
No. of cells per sq.in of Face Area	7.4	25.7
Width of Cell measured along Face in inches.	0.32 Ave.	0.16 Ave.
Length of Cell measured along Face in inches.	0.43 Ave.	0.24 Ave.

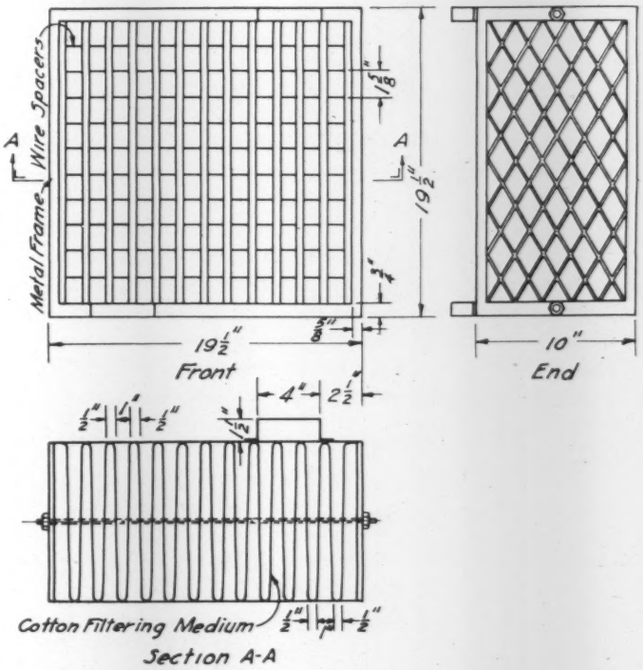
Fig. 3—Construction details of filter *C*

to drain for about 4 hours in an oven at 150 F. The properties of the filter oil are shown in Table 1. In order to expedite the work two of these filters were selected which had the same initial air resistance across the filter when cleaned but not oiled. After the filters were prepared for tests the air resistance across the filter was substantially the same. The slight differences were probably due to the difficulty of getting the filter absolutely clean after each test and covering it with a uniform film of oil. The differences in initial pressure were very slight and were considered to have no major influence on the test results.

Filter *B*, shown in Fig. 2, is a throw-away type made from viscous coated fibers graded as indicated in the sectional view. New filters were selected for each test.

Filter *C*, shown in Fig. 3, was a throw-away type made up of two layers of cells set at an angle to each other such that the air in passing through the filter must impinge on the sides of the cell. The size of the cells was graded to give better distribution of the cleaning effect. New filters were used for each test.

Filter *D*, shown in Fig. 4, was made up of a light cotton filter pad glazed on one side. The filter material appeared to be very uniform in texture and the assem-



Filtering Medium:-	Cotton - glazed on back side
Filtering Area:-	34.15 sq.ft.
Weight of Filtering Medium:-	9.85 grams per sq.ft.
Ratio of Area of Filtering Medium to Cross Sectional Area of Duct:-	12.3

Fig. 4—Construction details of filter *D*

Table 1—Properties of Viscous Coatings Used on Filters *A*, *B*, and *C*

Oil	Flash Point	Fire Point	Pour Point	Saybolt Universal Viscosity, Sec				Per Cent Ash
				70 F	100 F	130 F	210 F	
Filter *A*	505 F	575 F	10 F	4011	1222	486	93	1.50
Filter *B*	455 F	585 F	20 F	1170	426	230	—	2.64
					Furol Tube			
					100 F	122 F	210 F	
Filter *C*	560 F	600 F	60 F	—	2040	805	71	6.06

Table 2—Properties of Dusts Used in Tests

Dust	Density Grams per cc		Screen Anal. Per Cent by Weight		Moisture Loss Per Cent Heated to 175 F for 18 Hours
	Loose	Jolted to Max. Density	Between 200 and 325 Mesh	Pass 325 Mesh	
Fuller's Earth Passed 100 mesh.....	0.412	0.600	—	—	5.1
Fly Ash from Ill. Coal Passed 200 mesh.....	0.488	0.740	Approx. 23	Approx. 77	0.2
Pocahontas Ash Passed 200 mesh.....	0.455	0.766	Approx. 19	Approx. 81	0.15
Cottrell Ash Passed 100 mesh.....	0.888	1.285	—	—	0.98
Lampblack Passed 100 mesh.....	0.128	0.156	—	—	1.18
50 Per Cent Pocahontas 50 Per Cent Lampblack	0.215	0.316	—	—	—
50 Per Cent Pocahontas ash 20 Per Cent Ill. fly ash 20 Per Cent Lampblack 10 Per Cent Fuller's Earth	0.317	0.480	—	—	—

bled filter gave substantially the same initial resistance for each test. In this case one frame was used for all tests, the filter media being replaced between tests.

There was naturally some slight difference in the initial resistance of the various filters selected from each group. These differences were small and were considered to have no material effect on the final results of the tests.

Filters *A*, *B*, and *C* were of the viscous coated type and the properties of the coatings as determined in the laboratory are shown in Table 1.

Characteristics of Dust Used in Tests

Some of the properties of the dust used in the tests are shown in Table 2. The densities were determined under two conditions, first loose, and second jolted to maximum density. For the loose densities, measurements were made by pouring the dust into a container without jolting and for the maximum density the dust in the container was jolted until no further settling took place. As may be noted from the figures in the table, the ratio between maximum density and loose density varies from 1.22 for lampblack to 1.68 for Pocahontas ash, with an average of about 1.5 for all types used.

The density of the mixed dust as used in the tests was less than the weighted densities of the constituent parts. For instance, the maximum density of a mixture consisting of 50 per cent Pocahontas ash, 20 per cent lampblack, 20 per cent Illinois fly ash, and 10 per cent Fuller's earth gives a maximum density of 0.480 g per cubic centimeter, or 23 per cent less than the weighted density of the constituent parts, which was 0.622 g per cubic centimeter. The maximum density of a 50 per cent mixture of Pocahontas ash and lampblack was 0.316, or about 31 per cent less than the maximum weighted density of the constituent parts, which was 0.461 g per cubic centimeter. All of the different dusts were passed through the screen as indicated for the tests, and in addition the fly ash and Pocahontas ash were screened through a 325 mesh screen for purpose of analysis only. This screen passed 77 per cent of the fly ash and 75 per cent to 87 per cent of the Pocahontas ash. A 325 mesh screen will permit passage of particles having a minimum diameter in one direction of 43 microns. There was a considerable amount of variation in the results of this test on various samples of dust. As indicated in the last two columns, Fuller's earth showed the greatest moisture absorption with lampblack, and Cottrell ash next, and with Illinois fly ash and Pocahontas ash showing the least.

Test Apparatus and Procedure

With the exception of the dust feeding device the assembled test apparatus shown in Fig. 5 is the same as that specified in the A.S.H.V.E. Standard Code for Testing Air Cleaning Devices. The main test duct is 20 in. square and 9 ft long, connected by a reducing section 3 ft long and a 12 in. diameter section 5 ft long to an exhaust fan. The test filter is placed in the 20 in. square duct with the exhaust face about 8 in. from the reducing section. A direct current drive variable speed fan is used to draw the air through the filter, the volume being measured by a standard rounded orifice and the pressure drop across the filter being measured by a standard inclined water gage.

The dust feeding apparatus is shown at the left of Fig. 5 and consists of a revolving disc upon which the dust sample has been evenly distributed in a ring of uniform thickness. The disc containing the dust sample is placed on a rotating plate and a continuous uniform ribbon of dust is shaved from the outside edge of the dust ring by a specially designed knife edge fastened to the end of an air tube. Air is drawn through this tube by a low pressure air line and Venturi tube. The velocity of the entering air picks the ribbon of dust from the rotating plate and carries it through a tube to the distributing nozzle at the entrance of the filter test apparatus. The circular plate carrying the disc is rotated by a friction wheel in contact with the disc directly

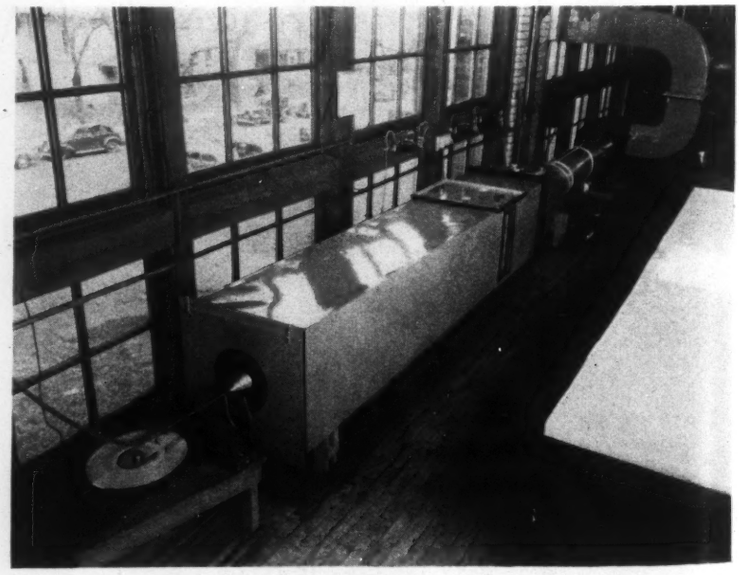


Fig. 5—Assembled view of test apparatus

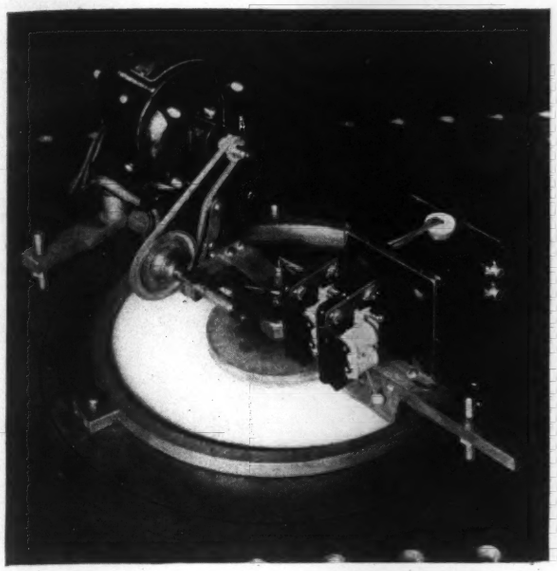


Fig. 6—Apparatus for distribution of dust sample on plate

under the dust pick-up tube. The rotating plate is moved forward by a screw mechanism which is operated at a constant relative speed to the rotating plate. Thus the ring of dust moves up toward the cutting knife at a constant speed, and since the tangential velocity of the disc under the cutting knife is maintained constant by the friction wheel underneath, the dust ribbon is of uniform cross section and there is a constant rate of dust feed to the filter.

In preparing the dust samples, aluminum discs similar in shape to phonograph records are used and placed in the dust distributing apparatus shown in Fig. 6. The disc is clamped down to a surface plate by an inner and outer ring. The required weight of dust is placed between these rings and leveled off by a rotating scraper or leveling device. With thin dust samples of less than about 60 g total weight it was necessary to use a vibrating apparatus in order to distribute the dust evenly over the plate. With thick samples of dust it is possible to omit this vibrating apparatus and still get uniform dust distribution.

In making a test the correct weight of dust for a one hour period is distributed on an aluminum disc. This disc is placed in the distributing apparatus, the rate of feed being determined by the speed of the disc. The volume of air passing through the filter is adjusted by the speed of the fan or by controlling the fan discharge outlet, and the air pressure drop across the filter is measured at intervals by a water gage. The amount of dust in the air leaving the filter is measured by drawing a certain percentage of this air through a fine grain alundum crucible for the continuous period of test and noting the gain in weight of the crucible due to the dust separated from the air. The arrestance calculations are based on the results of a one hour test. The dust arrestance for a one hour period is calculated as given:

$$\frac{\text{Weight of dust fed to filter} - \text{Weight of dust in air leaving filter}}{\text{Weight of dust fed to filter}} \times 100 = \text{arrestance per cent.}$$

The accuracy by which the amount of dust in the air leaving the filter could be measured by use of the alundum crucible was checked by running the test apparatus with no filter in the air stream. In these tests the amount of dust as measured by the crucible method showed a maximum variation of approximately 5 per cent from that actually fed into the filter, and the average variation was 1.6 per cent. For a filter of 90 per cent efficiency a variation of 5 per cent in the weight of dust leaving the filter would give a maximum possible error of 0.5 per cent in the over-all arrestance of the filter.

Procedure for Dust Concentration Tests

Filters *A*, *B*, *C*, and *D* were all used in this series. The dust mixture consisted of 50 per cent by weight of Pocahontas ash screened through a 200 mesh screen, 20 per cent by weight of Illinois fly ash screened through a 200 mesh screen, 20 per cent by weight of lampblack (Germantown Eaglebrand) screened through a 100 mesh screen, and 10 per cent by weight of Fuller's Earth, all thoroughly mixed and screened through a 100 mesh screen. A face air velocity of 300 fpm was selected, and the dust mixture was fed into the air stream at the rates of 10, 20, 30, 40, 60, and 100 g of dust per hour. This gave dust concentrations of 0.2, 0.4, 0.6, 0.8, 1.2 and 2.0 g per 1,000 cu ft of air respectively. For those filters in which it was evident from preliminary test data that small variations in the dust concentration would have no great effect on the filter arrestance, a part of the specified tests was omitted. In all of the tests the required amount of dust was fed into the air stream leading to the filter at a constant uniform rate, and the arrestances were taken at the end of each one hour period. On this basis each test was continued until the air pressure drop across the filter reached 0.4 in. of water.

The results of these tests are shown in Tables 3 to 8, inclusive, and in Figs. 7, 8, 9, and 10 for Filters *A*, *B*, *C*, and *D*, respectively. In order that the curves for each filter might be on a comparable basis the air resistances across the filter and percentage arrestance were plotted against weight of dust fed to the filter in grams. In each case the points only are shown for the arrestance curves at feeds of 10 g per hour. Due to the irregularity of these points the curves were omitted. This irregularity was due to the difficulty of determining the small increases in crucible weights necessarily accompanying a low rate of feed. Two crucibles were used alternately throughout all tests.

Referring to the figures in Table 3 and the curves of Fig. 7, which record the performance data for Filter *A*, it will be noted that there are slight variations in arrestance but no general trend to show an advantage for either the low or high rates of dust feed. The highest dust holding capacities were for the tests in which the rates of feed were 20 and 60 g per hour. The highest arrestance was for Test No. 3 at 40 g per hour. The dust holding capacity dropped off materially at a rate of feed of 100 g per hour, indicating that the filter was overloaded at this rate of feed. From 20 to 60 g per hour, however, the performance of the filter may be considered equally satisfactory for any rate. The dust catching surfaces of the filter were well graded throughout the filter and at rates of dust feed below 60 g per hour there did not seem to be a tendency to overload the filter.

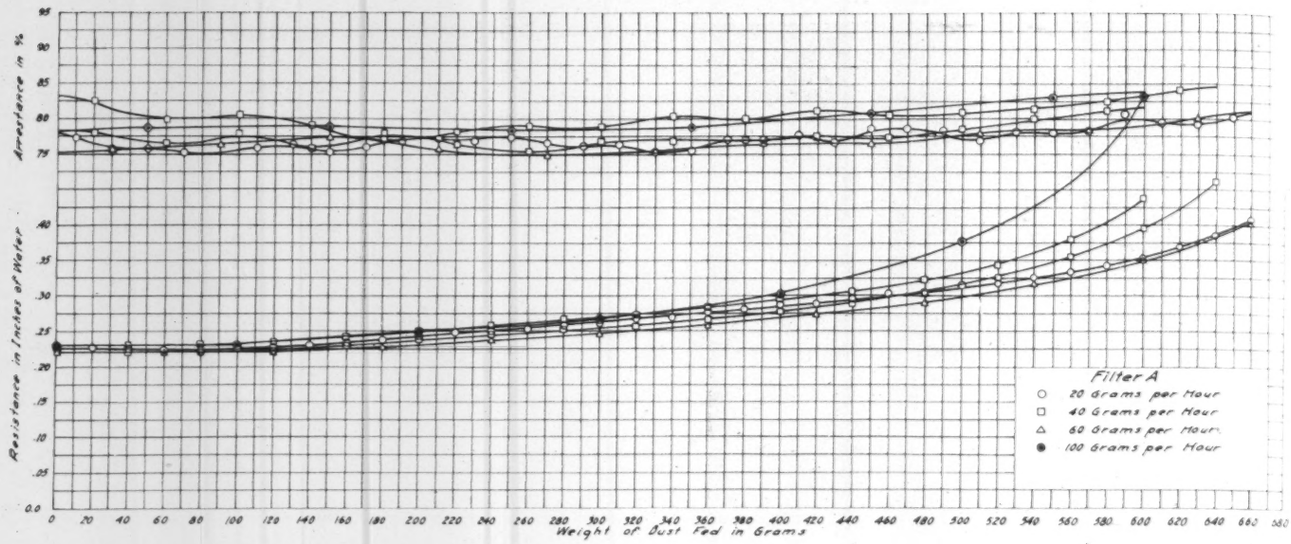

Arrestance in %
Resistance in Inches of Water
Weight of Dust Fed in Grams
Filter A
20 Grams per Hour
40 Grams per Hour
60 Grams per Hour
100 Grams per Hour

Fig. 7—Arrestance and resistance curves at various rates of dust feed for filter *A*

Table 3—Filter *A*. Test Results for Different Dust Concentrations
(*Face Air Velocity* 300 *fpm. Dust mixture* 50, 20, 20, 10)

Test No.	Rate of Feed	Initial Resis.	Length of Test	Gram-Hours	Avg. Arrest-ance	Dust Holding Capacity
16	20	0.225	32.8	656	77.2	503.8
134	40	0.229	14.5	580	77.5	449.0
111	40	0.220	15.1	604	80.3	483.6
112	60	0.220	11.0	657	76.9	504.8
144	100	0.230	5.2	520	79.7	411.9

Table 4—Filter *B*. Test Results for Different Dust Concentrations
(*Face Air Velocity* 300 *fpm. Dust mixture* 50, 20, 20, 10)

Test No.	Rate of Feed	Initial Resis.	Length of Test	Gram-Hours	Avg. Arrest-ance	Dust Holding Capacity
106	10	0.146	24.8	248	80.0	199.8
12	20	0.153	15.8	316	78.8	249.2
102	30	0.150	9.6	288	81.3	233.6
104	40	0.148	8.0	320	81.0	259.2
103	40	0.152	7.3	292	81.9	237.5
101	40	0.150	8.7	347	81.7	282.4
107	60	0.157	5.4	324	82.0	264.6
143	100	0.147	3.9	390	82.8	322.8

Table 5—Filter *C*. Test Results for Different Dust Concentrations
(*Face Air Velocity* 300 *fpm. Dust Mixture* 50, 20, 20, 10)

Test No.	Rate of Feed	Initial Resis.	Length of Test	Gram-Hours	Avg. Arrest-ance	Dust Holding Capacity
145	10	0.142	29.0	290	72.3	210.7
11	20	0.148	15.6	312	72.0	221.0
105	40	0.142	9.0	358	69.9	250.3
127	60	0.156	6.0	360	71.4	256.0
142	100	0.147	4.6	460	68.2	313.9

Table 6—Filter *D*. Test Results for Different Dust Concentrations
(*Face Air Velocity* 300 *fpm. Dust Mixture* 50, 20, 20, 10)

Test No.	Rate of Feed	Initial Resis.	Length of Test	Gram-Hours	Avg. Arrest-ance	Dust Holding Capacity
110	10	0.086	20.7	207	89.8	186.2
31	20	0.088	10.8	216	92.0	198.5
113	20	0.091	11.4	228	92.5	208.8
108	40	0.095	6.3	252	93.5	233.5
109	60	0.090	4.1	246	94.6	231.1
149	100	0.087	2.4	240	92.4	222.5

Table 7—Average Dust Arrestances for Filters *A*, *B*, *C*, and *D*
(*Face Air Velocity* 300 *fpm. Dust Mixture* 50, 20, 20, 10)

Filter	Rate of Dust Feed—Grams per Hour					
	10	20	30	40	60	100
(*A*)	—	77.2	—	77.5 80.3	76.9	79.7
(*B*)	80.0	78.8	81.3	81.0 81.9 81.7	82.0	82.8
(*C*)	72.3	72.0	—	69.9	71.4	68.2
(*D*)	89.8	92.5 92.0	—	93.5	94.6	92.4

Table 8—Average Dust Holding Capacity up to 0.4 in. Pressure Drop for Filters *A*, *B*, *C*, and *D*
(*Face Air Velocity* 300 *fpm. Dust Mixture* 50, 20, 20, 10)

Filter	Rate of Dust Feed—Grams per Hour					
	10	20	30	40	60	100
(*A*)	—	503.8	—	449.0 483.6	504.8	411.9
(*B*)	199.8	249.2	233.6	259.2 237.5 282.4	264.6	322.8
(*C*)	210.7	221.0	—	250.3	256.0	313.9
(*D*)	186.2	198.5 208.8	—	233.5	231.1	222.5

The test results for Filter *B* are recorded in the figures of Table 4 and by the curves of Fig. 8. From these results it would appear that the performance of the filter improves as the rate of dust feed is increased. However, there is a variation in the test results for different filters under the same test conditions as noted in Tests Nos. 104, 103, and 101, all of which were run at a dust concentration of 40 g per hour on filters which had substantially the same initial resistance. In general, the performance at rates of feed from 30 to 60 g of dust per hour checked within reasonable limits. The low dust holding capacity at 10 g and high dust holding capacity at 100 g would indicate a better distribution of dust throughout the filter at the high rates of feed than at the very low rates of feed.

The test results for Filter *C* are shown in the figures

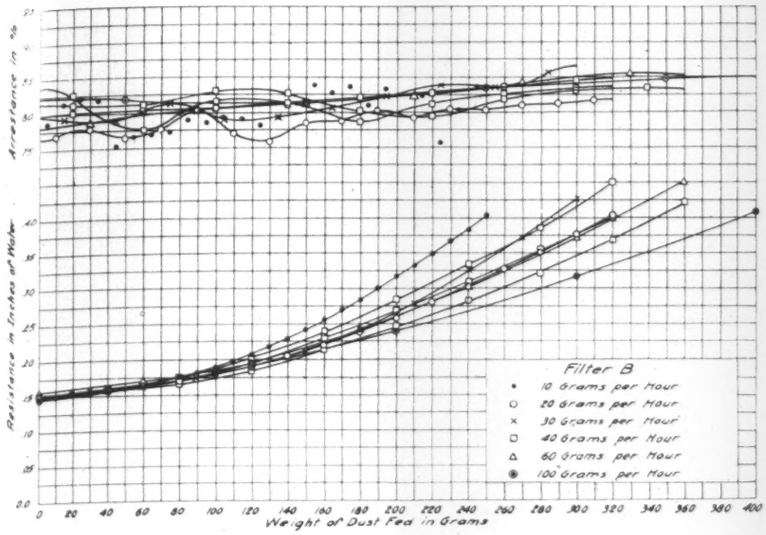

Filter B
10 Grams per Hour
20 Grams per Hour
30 Grams per Hour
40 Grams per Hour
60 Grams per Hour
100 Grams per Hour
Weight of Dust Fed in Grams

Fig. 8—Arrestance and resistance curves at various rates of dust feed for filter *B*

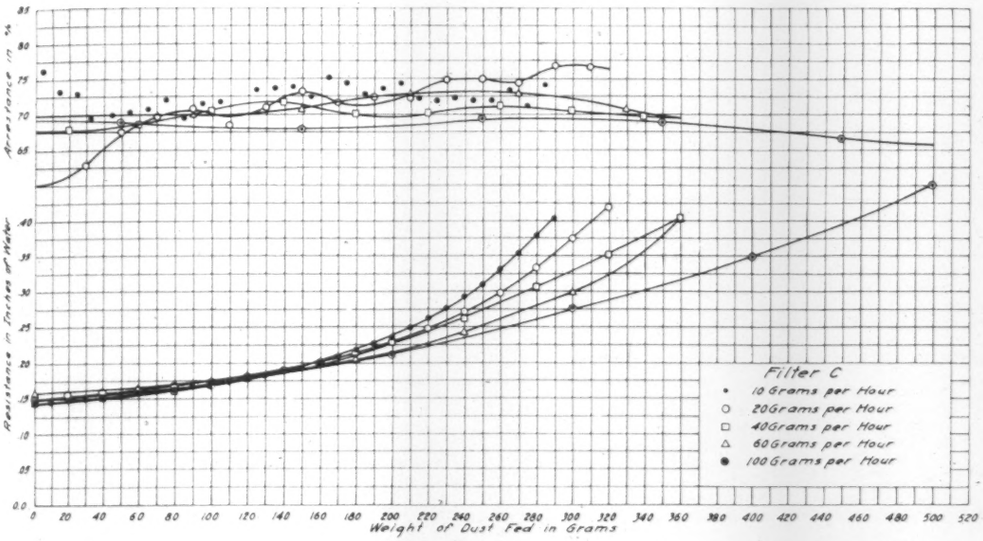

Filter C
10 Grams per Hour
20 Grams per Hour
40 Grams per Hour
60 Grams per Hour
100 Grams per Hour
Weight of Dust Fed in Grams

Fig. 9—Arrestance and resistance curves at various rates of dust feed for filter *C*

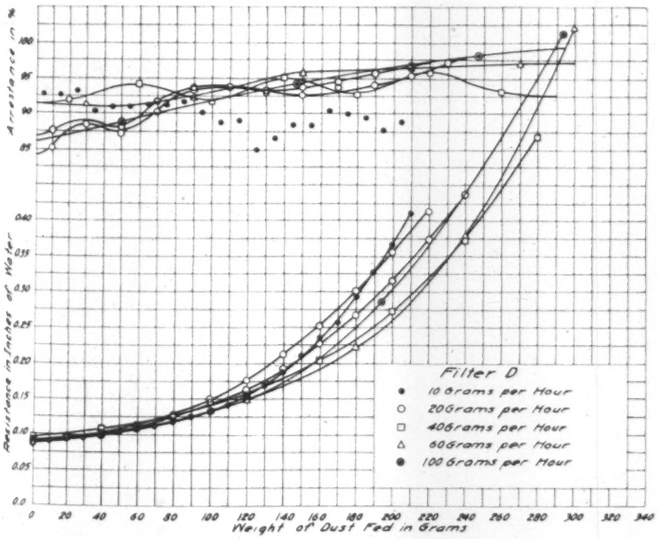

Filter D
10 Grams per Hour
20 Grams per Hour
40 Grams per Hour
60 Grams per Hour
100 Grams per Hour
Weight of Dust Fed in Grams

Fig. 10—Arrestance and resistance curves at various rates of dust feed for filter *D*

of Table 5 and the curves of Fig. 9. In this filter, as in Filter *B*, the general performance characteristics tend to improve with the higher rates of dust feed. The dust holding capacity increases directly with the rate of feed, and while there is a slight tendency for a reduction in dust arrestance at the higher rates, this reduction is not great and the average arrestance only drops from 72.3 per cent at 10 g of dust per hour to 68.2 per cent at 100 g of dust per hour. The dust holding capacities for the low and the high rates of feed are substantially the same for Filters *B* and *C*.

The test results for Filter *D* are shown in the figures of Table 6 and the curves of Fig. 10. For this filter the dust holding capacity increased with rate of dust feed from 10 to 40 g per hour with approximately the same capacity at 60 g per hour. At 100 g per hour there is a tendency toward a decrease in dust holding capacity. Both the arrestance and dust holding capacity are lower for the 10 g rate than for the higher rates. From 20 to 100 g of dust per hour the arrestances are substantially the same, the highest being at 60 g per hour.

The conclusions which may be drawn from this series of tests are as follows:

1. Within the range of 10 to 100 g of dust feed per hour the rate of feed has no appreciable effect on the arrestance of any of the four types of filters tested. This conclusion is borne out by reference to Table 7 which shows a summary of the average arrestances for all filters at different rates of feed. The spread between the average arrestance values for any given filter at different rates of feed might have been reduced if more tests had been used to determine the average value for each rate of dust feed.

2. As shown in Table 8 there was but very little change in dust holding capacities in any of the filters due to varying the rate of dust feed from 20 to 60 g per hour. For Filters *D* and *C* there was a marked increase in dust holding capacities at a rate of 100 g per hour, and for Filters *B*, *C* and *D*, the only ones tested at 10 g of dust per hour, there was a decrease in dust holding capacity for the 10 g rate. For Filters *B* and *C* the dust holding capacity at 100 g of dust feed per hour was approximately 50 per cent greater than the dust holding capacity for 10 g of dust per hour. For Filter *A* the dust holding capacity at 100 g per hour is a considerable amount lower than that for any of the feeds between 20 and 60 g per hour. The increase in dust holding capacity for Filters *B* and *C* at 100 g rate indicated that at this high rate of feed a lesser amount of dust was retained on the entering surface of the filter and therefore more dust was carried back into the filter to give a more uniform distribution throughout the filter volume, thus making it possible for the

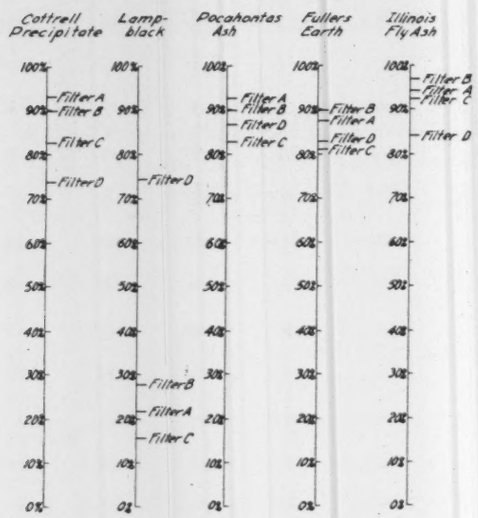


Fig. 11 — Comparative filter arrestances for component dusts

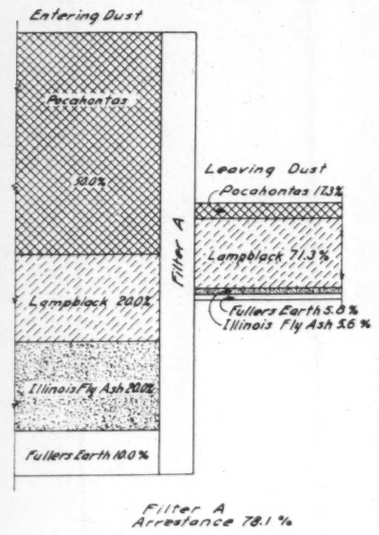


Fig. 12—Relative weights of mixed dust entering and leaving filter *A*

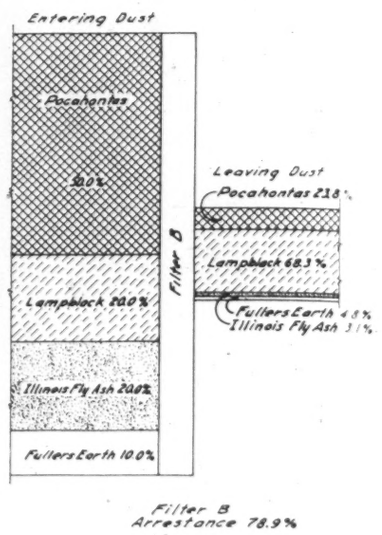


Fig. 13—Relative weights of mixed dust entering and leaving filter *B*

filters to retain more dust without building up an excessive air resistance across the filter.

3. Insofar as dust arrestance is concerned, the rate of dust feed for standard air filter rating tests might be set at any value from 10 to 100 g per hour, and insofar as dust holding capacity is concerned it might be set at any value from 20 to 60 g per hour with substantially the same results for any filters of the types tested.

Procedure for Air Velocity Tests

In this series the four Filters *A*, *B*, *C*, and *D* were tested. The dust mixture used was the same as in the previous series and it was fed into the air stream at the rate of 20 g per hour. Each filter was tested at air velocities of 200, 300, and 400 fpm. The tests were all three hours in length, the arrestance and air resistance being determined for each hour of the test. Since the purpose of the test was to determine the effect of air velocity on the dust arrestance of the filter, it was not considered necessary to run the test for more than a three hour period.

A summary of the test results is shown in Table 9. These tests show that there is but very little difference in the dust arrestance between the air velocity of 200 and 400 fpm. There is, however, in all cases a slight increase at the 400 fpm air velocity, and in Filter *A* this increase is practically uniform over the range from 200 to 400 fpm air velocity. In Filters *B*, *C*, and *D* the dust arrestance at 300 ft is slightly less than that at 200 fpm air velocity. The increase in arrestance over the range of tests runs from 1.2 per cent to 5.8 per cent for Filters *C* and *D* respectively.

Even though these increases are small they indicate that velocities higher than 300 fpm might be used on any of the filters without impairing their dust catching arrestance.

Table 9—Dust Arrestance at Different Air Velocities
(*Dust Mixture* 50, 20, 20, 10. *Dust feed* 20 *g per hour*)

Air Velocity in Fpm	Filter			
	A	*B*	*C*	*D*
200	74.4	77.4	65.7	89.1
300	76.1	77.0	65.0	86.9 87.8
400	77.6	78.7	66.2	92.6
Range (Per Cent Diff.)	3.2	1.7	1.2	5.8

Filter Characteristics as Governed by Constituent Dusts of Mixture

The purpose of this series of tests was to determine the effect of individual dusts used in the dust mixture on the operating characteristics of the filter and to compare the arrestance values as determined by tests of filters when using various dust mixtures with the weighted values as obtained when using individual dusts of the mixture. In this series of tests all filters were used, the air velocity through the filter was set at 300 fpm, and the dust mixtures were fed to the filters at the rate of 20 g per hour. All tests were four hours in length excepting in those cases where the filter resistance increased to 0.4 in. of water before the end of the four hour period. The dust arrestance and air resistance were recorded for each one hour period. The results for these tests are shown in Tables 10, 11, and 12, and in Figs. 11 to 15 inclusive.

The average arrestance values and increase in resistance across the filter for five different types of dust are shown in Table 10. The increase in air resistance is shown for the three hour period, and for all types of dust excepting lampblack the increase is small. In the case of Filter *A* there was a slight decrease in resistance when using Illinois fly ash and Fuller's Earth. This was probably due to readjustment of oil on the filter media and the streamlining of the oil on the fibers due to passage of air during initial stages of test. The rise in resistance when using lampblack was high in all cases, but this is particularly marked for Filter *D* in which the three hour increase was recorded as 0.288 in. of water. The dust arrestance for Filter *D* when using lampblack was much higher than for any of the other filters. This partly accounts for the rapid increase in air resistance of Filter *D* as compared with the other filters.

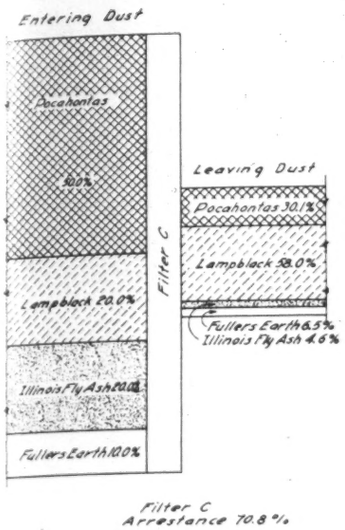


Fig. 14—Relative weights of mixed dust entering and leaving filter *C*

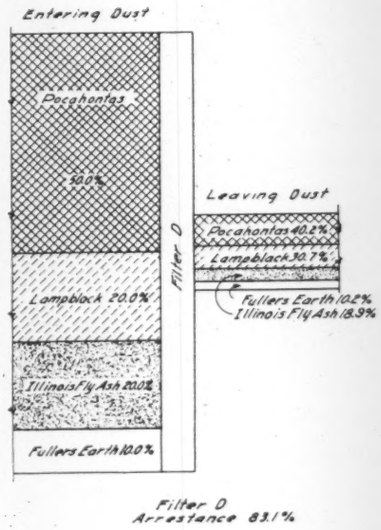


Fig. 15—Relative weights of mixed dust entering and leaving filter *D*

Table 10—Average Dust Arrestance and Increase in Air Resistance for Different Types of Filters[a]

(Dust fed at rate of 20 g per hour for a three hour period)

Type Dust	Filter *A*		Filter *B*		Filter *C*		Filter *D*	
	Resist. Increase In. of H_2O	Average Arrestance	Resist. Increase In. of H_2O	Average Arrestance	Resist. Increase In. of H_2O	Average Arrestance	Resist. Increase In. of H_2O	Average Arrestance
Illinois Fly Ash	—0.001	94.0	0.011	96.7	0.003	92.1	0.004	84.0
Pocahontas Ash	0.001	92.4	0.009	89.8	0.008	82.5	0.006	86.4
Cottrell Ash	0.001	92.8	0.004	89.9	0.002	82.3	0.006	73.7
Fuller's Earth	—0.004	87.2	0.011	89.8	0.002	81.0	0.006	82.9
Lampblack	0.002	21.7	0.023	27.8	0.022	15.3	0.288	74.1

[a]Average Arrestance and Resistance Increase for three hour period.

Table 11—Spread Between Dust Arrestance Values for Different Kinds of Dust

(Air Velocity 300 fpm. 20 g dust per hour)

Type of Dust	Efficiency Spread		
	High	Low	Per Cent Spread
Illinois Fly Ash	96.7 *B*	84.0 *D*	12.7
Fuller's Earth	89.8 *B*	81.0 *C*	8.8
Pocahontas Ash	92.4 *A*	82.5 *C*	9.9
Lampblack	74.1 *D*	15.3 *C*	58.8
Cottrell Ash	92.8 *A*	73.7 *D*	19.1

Table 12—Comparisons of Arrestance Values as Determined by Tests with Dust Mixtures and by Weighted Values from Tests on Individual Dusts

(Air Velocity 300 fpm. 20 g dust per hour)

Type of Filter	50% Pocah., 20% Ill. 20% Lamp., 10% Full.		50% Pocahontas 50% Lampblack		Per Cent Difference	
	Straight Mix Test	Weighted Values	Straight Mix Test	Weighted Values	50–20–20–10	50–50
(*A*)	76.1	78.1	54.6	57.0	+2.0	+2.4
(*B*)	77.0	78.9	56.2	58.9	+1.9	+2.7
(*C*)	65.0	70.8	46.5	48.9	+5.8	+2.4
(*D*)	87.7 86.9	83.1	81.5	80.3	—4.6 —3.8	—1.2

The spread in arrestance values for the different types of dust are shown in the figures of Table 11 and in the chart of Fig. 11. The three greatest spreads were 58.8 per cent for lampblack, 19.1 for Cottrell precipitate, and 12.7 per cent for Illinois fly ash. In each of these cases the wide range is caused mostly by the characteristics of Filter *D*. This is clearly shown by the chart of Fig. 11. Filter *D* is not quite as efficient on Cottrell precipitate and Illinois fly ash as the other filters, but is very much more efficient on lampblack. The standard lampblack as used consists of very fine particles which are difficult to remove by the ordinary viscous coated filters. It seems to be removed only by the very fine mesh part of the filter, and when so removed builds up the air resistance in the filter very rapidly. The large percentage (50 per cent) of this material specified in the A.S.H.V.E. Standard Test Code and used in former tests undoubtedly accounts for the low arrestance values obtained for some types of filters.

The arrestance values for all filters were determined for the Standard Code mixture of dust consisting of 50 per cent Pocahontas ash and 50 per cent lampblack. These values were also determined at air velocities of 300 fpm and for 20 g of dust fed to the filter per hour. The arrestance values for these tests and the arrestance values for the 50 per cent, 20 per cent, 20 per cent, 10 per cent dust mixture tests were compared with the weighted arrestance values as calculated from the test values for each individual dust on the same filter. The weighted values were calculated by multiplying the test values for each separate dust by the percentage of that dust in the final mixture and adding the results for all dusts of the mixture. These comparative values are shown in Table 12. It is interesting to note that in general there is a very close agreement between the calculated weighted values and the test values for the different dust mixtures. This means that any dust in the mixture will affect the performance characteristics of the filter in direct proportion to the percentage of that dust in the mixture. Since the lampblack which has been in common use for testing filters according to the Standard Code is a very difficult dust for most filters to remove, and since, when efficiently removed, it builds up the filter resistance very rapidly, it is evident that the large percentage specified in the Standard Test Code severely handicaps the performance of most filters either by lowering their dust arrestance values or by increasing the air resistance values and thus shortening the life of the filter.

Figs. 12, 13, 14, and 15 show in diagram form the performance characteristics of Filters *A*, *B*, *C*, and *D* when using the dust mixtures indicated. The percentage values shown for the dust before and after passing through the filter indicate the percentages of the individual type of dust in the initial and final weights of dust. These diagrams show very clearly the effect of different types of dust in the mixture on the arrestance of a given filter. For instance, for Filters *A*, *B*, and *C* there is a large percentage of lampblack in the final dust.

This explains the low arrestance values for the 50 per cent mixture as shown in Table 12. In Filter *D* there is a small percentage of lampblack in the final dust, thus explaining the high arrestance for this filter as shown in Table 12. The resistance, however, as shown by Table 10, would be built up very rapidly by an excess of lampblack.

Conclusions

The final conclusions from this investigation which may be applied to the Test Code are as follows:

1. For rating purposes it would be practical to use heavier dust concentrations than now specified by the Code. From the first series of tests it is apparent that dust concentrations from 0.6 to 1.2 g of dust per 1,000 cu ft of air would give test results which would be equally as valuable as those at dust concentrations of 0.35 g per 1,000 cu ft as specified in the Code. Higher concentrations would greatly expedite any tests to determine the life of a filter. There seems to be some question as to the arrestance, and particularly the life of a filter at very low dust rates. However, any tests to determine the life of a filter at a rate of dust feed comparable to that which may be met in practice would require unlimited time and would be impractical.

2. The air velocity at which a filter is tested does not appear to be important, providing it is maintained between 200 and 400 fpm. Further investigations should be made if other velocities are to be used. There seems to be no practical reason for changing the 300 fpm velocity as specified in the present Code, as this would not particularly expedite the tests.

3. Lampblack is a very difficult dust to remove from the air and, if mixed in the original dust in large quantities, it will unduly handicap the performance of most filters, either by reducing the dust arrestance value or by increasing the air resistance and thus shortening the life of the filter. It is believed that 50 per cent of lampblack is very much in excess of that found in practice and it would seem reasonable to reduce this percentage in the Standard Test Code.

4. The performance of a filter varies greatly with the mixture of dust used. By proper choice of a test dust, one type of filter may be shown to have better performance characteristics than the majority of the other filters, whereas another dust mixture may reverse the relative merits. Three of the four filters tested ranked first in arrestance on one particular dust. This does not infer the fourth filter to be inferior in dust arrestance, as further tests on this same type of filter but with a different design, brought its arrestance up among the highest on one particular dust mixture.

These tests were made for the purpose of studying the characteristics of different types of filters when rated by a laboratory test method and to determine whether or not some improvements might be made in the Standard A.S.H.V.E. Test Code requirement. When interpreting the test results it should be remembered that only one design of each type of filter was selected, and that the filters were not all of the same thickness, nor did they represent the only thickness manufactured for the different types. Other designs of some of the types might have been selected which would have changed the relative merits of certain properties of the filters as shown by the test results. To compare the merits of the different types of filters would require a much more extended series of tests and any attempt to make comparisons from the data presented here would be misleading due to the limited selection from each type. The selections were, however, sufficiently broad to substantiate the recommendations made in regard to changes in test procedure and Code.

There is a need for further investigation along the following lines:

1. The performance of different types of filters should be investigated when using different types of dusts and dusts of different particle size.

2. A further analysis should be made covering the different types of carbon which fulfill the Standard Code requirements.

3. A study should be made to determine the effect of lint on the performance of various types of filters.

4. A more complete analysis should be made of the dust which must be handled in different types of filter installations. From this analysis it may be found necessary to build up more than one type of synthetic dust for adequately rating different types of filters for different classes of installations.

5. A practical field test should be devised which could be used in the field and which could be used parallel with the laboratory rating test.

Effect of Reactivity and Caking When Burning Coal

The relative rates of burning of coals when all other conditions, such as size of the pieces, depth of bed, rate of flow of air through the bed, and the furnace, are the same, depend mainly on the effect of the caking of the coal on the distribution of the air and on the relative area of coal the air can contact. The rate of burning of the coke which is formed from the coal in the fuel bed depends on the size of the pieces of coke and on their reactivities, that is, relative ability of the carbon of the coke to unite with oxygen, carbon dioxide, and water vapor. These facts were emphasized recently by Percy Nicholls, heat transmission engineer, Bureau of Mines, Department of the Interior, in presenting the results of some experimental studies of fuel beds before the Fuel Engineering Division of Appalachian Coals, Inc., at Cincinnati, Ohio.

Measurements of reactivities usually have been associated with manufactured cokes, and little is known about the reactivities of cokes formed in fuel beds. However, these studies showed that although values for reactivity of cokes vary largely at temperatures below 2000 F, differences between these values decrease as the temperature increases and will be negligible at the temperatures which exist in industrial fuel beds. On the other hand, decrease in the size of the pieces of coke increases the rate of gasification independently of temperature and, therefore, the size of the pieces materially affects the rate of burning of fuel beds which are similar in all other respects, including the rate of flow of air through them.

Caking affected both the initial resistances of the fuel beds and the average rate of burning. The test results do not mean that high rates of burning cannot be obtained with strongly caking coals, or that caking has not offsetting advantages. They do show, according to Mr. Nicholls, that more knowledge is required on the effect of caking in fuel beds, both for the design of equipment and in the selection of coal for existing plants.

A Test Method for Air Filters*

By Richard S. Dill**, Washington, D. C.

THE apparatus now used for testing air filters at the National Bureau of Standards consists of a means of drawing air through a specimen filter at a predetermined rate, a means of impregnating the air with dust before it reaches the filter under test and a means of drawing measured samples of the filtered and unfiltered air simultaneously through filter papers. A photometer which employs a source of light and a photoelectric cell is provided for comparing the densities of the dust deposits on the filter papers. At present the standard test dust is a fly ash called Cottrell precipitate since it is obtained from a Cottrell precipitator in a local power plant burning pulverized coal. Whatmans No. 41 filter paper has been found satisfactory and has been used during the tests.

The general arrangement of the apparatus is shown in Fig. 1. The air ducts shown and the elutriating tank are made of 24 gage galvanized iron. Air enters through the duct connecting to the window, as indicated at the top right of Fig. 1, and is discharged from the lower pipe through the same window.

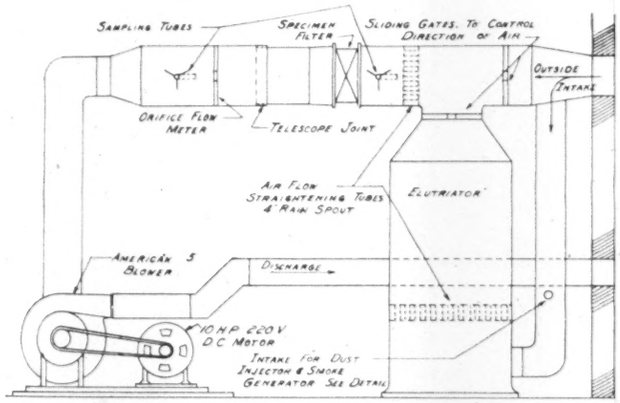


Fig. 1—Diagram of testing equipment for air filters

The entering air can be passed directly to the filter under test, as was formerly done for trials with atmospheric dust, or it can be shunted downward through the 10-in. pipe shown and forced to come upward through the elutriating tank before reaching the specimen.

The purpose of the elutriating tank is to arrest abnormally large particles and prevent them from reaching the filter.

There are flow straighteners consisting of nests of 4-in. gutter pipe in the elutriating tank and in the horizontal duct between it and the filter.

Leaving the elutriating tank the air passes through the specimen filter and the metering orifice, goes down through a vertical pipe, passes through the blower and is ejected out of doors through the lower duct.

A telescoping section in the main duct permits filters up to about 30 in. in length to be accommodated. Since filters differ in size and shape a wooden or sheet metal duct section is usually constructed to receive the filter and the duct section is fitted in the space provided in the main duct.

A slant gage is used to measure the pressure loss or resistance of the filter. A U-tube water manometer is used to indicate the pressure drop across the metering orifice and hence the air flow in the main duct. A 6-in. orifice is used for deliveries up to about 1200 cfm and a 9-in. orifice for flows up to 2000 cfm which is near the upper limit of capacity of the apparatus

For taking samples of air to be passed through filter papers, sampling tubes are used, one on the clean air and one on the dirty air side of the filter under test. Their positions are indicated on Fig. 1 and the construction of the sample collector is shown on Fig. 2.

Each collector is made of 1½-in. brass tubing and is about 12-in. long. It contains a metering orifice about 0.18-in. in diameter. It is reduced in size and fitted with a half-inch standard pipe connection at the entrance end. The discharge end is covered by a cap held against the flat end of the tube by a pair of springs. The springs can be stretched by hand to permit a filter paper clasp to be inserted between the cap and tube. The cap is connected by a rubber tube to a vacuum line. The filter paper clasp consists of two matching brass plates with an aperture across which the filter paper is stretched.

Each sampling tube consists of a 2-ft length of half-inch pipe which enters the main duct at right angles and turns on a 6-in. radius to face up-stream. Air is drawn from the main duct through the pipe and into the collector. In the collector the air passes first through the metering orifice, then through the filter paper and is discharged to the vacuum line.

A one-leg water manometer is used to indicate the air flow rate through the metering orifice in each collector. The manometers for the two collectors are mounted ad-

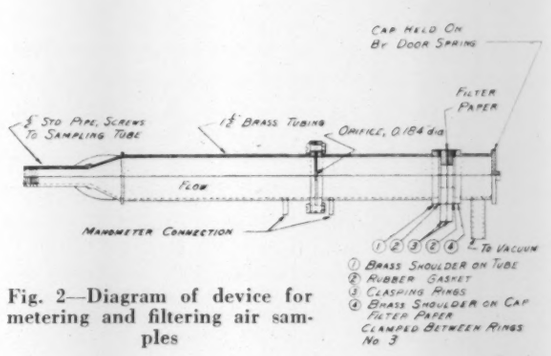

Cap Held On By Door Spring
Filter Paper
½ Std. Pipe, Screws To Sampling Tube
½ Brass Tubing
Orifice, 0.184 dia.
Flow
Manometer Connection
1 2 3 4 5 To Vacuum
1 Brass Shoulder on Tube
2 Rubber Gasket
3 Clashing Rings
4 Brass Shoulder on Cap
Filter Paper Clamped Between Rings No. 3

Fig. 2—Diagram of device for metering and filtering air samples

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jacent to each other for convenience and their scales are graduated to read air flow in cubic feet per minute. Their readings are called sampling rates and the sampling rates can be adjusted by needle valves in the vacuum lines.

The photometer used for comparing the density of dust deposits on filter papers consists essentially of an automobile headlight bulb and a photo-cell with an aperture across which the filter paper can be placed so that the light from the bulb passes through the paper and falls on the cell. The aperture is, of course, smaller than any dust spot taken. The arrangement is shown on Fig. 3.

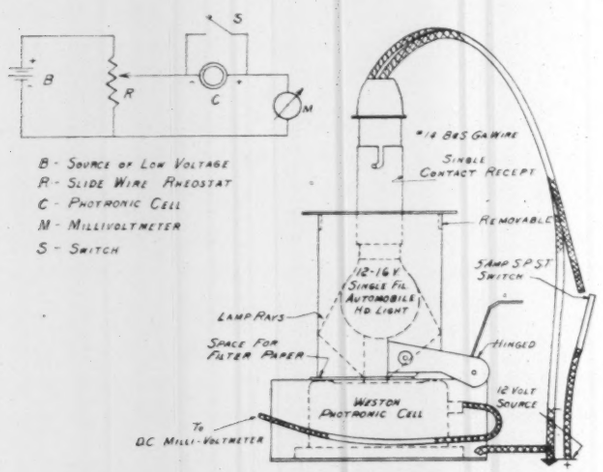

Fig. 3—Diagram of photometer for measuring light transmission of filter papers

Originally the cell was connected directly to a galvanometer and the reading of the galvanometer was considered indicative of the intensity of the light striking the cell. This arrangement was improved by employing a circuit in which the effect on the cell of external resistance is made unimportant. The circuit is described in a paper by Wood.[1] The wiring diagram is shown on Fig. 3. In use, the source of low voltage acts in the same direction in producing current in the circuit as the photo-cell. The rheostat, R_1, is adjusted until opening and closing the switch makes no difference in the reading of the millivoltmeter. For this condition the cell is acting against no resistance external to itself so that the effect of changes in its internal conductivity is avoided. The readings of the millivoltmeter at this condition are accepted as an indication of the light passing through a filter paper and falling on the cell and, consequently, of the transparency of the paper.

The device for impregnating air with dust is shown in Fig. 4. It is located on the 10-in. pipe ahead of the elutriating tank in the system. The elements of the device are a drum, rotated slowly by a motor, a dust hopper above the drum arranged to lay a ribbon of dust on it, and an aspirator, operated by compressed air, arranged to draw the dust from the drum and inject it into the 10-in. pipe in which the main air stream is passing on its way to the elutriating tank and filter. The dust injector is regulated to pass about one gram of dust per thousand cubic feet of air. Ten grams are charged into the hopper for a run so that the duration of a run for a filter with a delivery of 800 or 1000 cfm is 10 or 12 min.

The use of this quantity of dust at this rate has been found to result in dust spots on the filter papers of good density for convenient comparison.

Air is drawn through a filter mounted for test at the rate specified for it by its maker. Its resistance is noted and trials of its efficiency are made at this rate.

Present practice during trials for efficiency is to draw samples of air at equal rates, usually 1 cfm, through the upstream and downstream filter papers. The spot on the downstream paper exposed for the passage of air and hence for dust deposition is smaller than that on the upstream paper. If the area of the downstream spot is half that of the upstream spot and the spots are found to be equally dense, this means that the upstream air carried twice as much dust as the downstream air and the efficiency by this test method is therefore considered to be 50 per cent. If the downstream spot is found to be the cleaner, the efficiency is considered to be above 50 per cent, and vice versa.

The size of a filter paper spot is governed by the size of the aperture in the filter paper clasp. Pairs of clasps are provided having apertures corresponding to efficiencies of 50, 60, 65, 70, 75 and 80 per cent. The present routine test consists of finding by trial that the efficiency of a specimen filter is above one of these values and below the next higher.

Evaluation of an efficiency with a precision of one or two per cent is possible. Such evaluations are not customarily made because integral efficiencies, multiples of five, have been chosen by the purchasing agencies of the Government as criteria for the performance of the various classes of filters. Filters with efficiencies below 60 per cent are not acceptable and filters with efficiencies above 80 per cent are generally acceptable under present Government specifications.

In preparation for a run all parts of the apparatus are swept out by drawing air through them at a comparatively high rate. A pair of filter papers, previously found to be photometrically similar, is placed in a pair of clasps and these are placed in the spaces provided in the sample collectors. Ten grams of dust are put in the dust injector. Flow rates through the filter and through the sampling tubes are adjusted and the dust injector is then started. The run continues until all the dust has been injected, at which time the flow through the sampling tubes is cut off and the papers are removed for photometric examination. Successive runs are made until the efficiency is evaluated as indicated above.

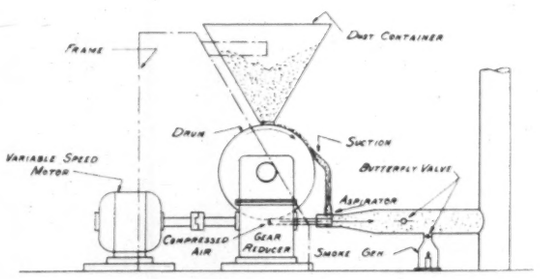

Fig. 4—Diagram of dust injector and smoke generator

[1]Zero-Potential Circuit for Blocking-Layer Photo-Cells, by Lawrence A. Wood (*The Review of Scientific Instruments*, Vol. 7, No. 3, p. 157, March, 1936).

The apparatus was installed at the Bureau in 1934 at the request of the Procurement Division of the Treasury Department and it has been used since that time without essential change as a type test for air cleaning devices offered for sale to the Government.

At the outset, tests of filters with atmospheric dust, that is dust normally in the outside air on a clear day at the Bureau of Standards, were attempted. It was found, however, that the efficiencies of filters were uniformly low and difficult to evaluate with this dust. Trials with smoke from a fuel oil flame showed that this substance was subject to the same objections, and therefore tests with these two dusts are no longer commonly made.

With the Cottrell precipitate, however, efficiencies of filters on the market were found to be spread over a wide range, specifically from less than 10 to something over 80 per cent and the differences between filters are more easily shown.

A variation of the test method has been used as a basis for a specification for electro-static dust precipitating equipment. The specification required that the equipment have an efficiency of 90 per cent or more on atmospheric dust on any clear day. The specified test consisted of drawing air at equal sampling rates through filter paper spots of equal area on the two sides of the equipment, and drawing air through the downstream spot ten times as long as through the upstream spot. The requirement was that under this condition the downstream spot should not be the denser.

Means of improving the test are being sought. Some efforts to separate the dust into component size ranges by elutriation have shown promise but have not been carried to a conclusion.

An efficiency test apparatus of different design has been tried, also with promising results. This apparatus consists essentially of a means of drawing samples of air through two filter papers simultaneously while light from a single bulb passes through the two papers and falls on two photo-cells.

The cells are connected to the same galvanometer and if the light falling on one cell decreases faster than the light falling on the other, the galvanometer moves off the zero position. In making a test, the flows of air through the filter papers are kept in such ratio as to keep the galvanometer on the zero position, showing that the papers are becoming dirty at equal rates. The ratio of the sampling rates is then an index to the efficiency of the filter under test. With this apparatus a continuous record of the efficiency of a filter over a period of time is possible.

The test method as it is now applied gives information on the performance of filters when new. It gives no information on the effects of time or use on performance but its application has permitted the selection of filters with useful initial effectiveness and this, in considerable degree, puts the selection of filters on a definite basis. The resistance of filters commonly increases with dust load and it is probably in order for the test to be extended or altered to obtain information on this point. However, such information is held to be of secondary importance compared to knowledge of the ability of filters to arrest dust, since arresting dust is the prime purpose of a filter.

Manufacture and Use of Safety Glass Shown in Educational Film

The manufacture and utilization of safety glass is depicted in a 2-reel silent motion picture film entitled, Safety Glass, which has been added to the educational film library of the Bureau of Mines, United States Department of the Interior.

The measure of protection afforded to life and property by the use of safety glass in automobiles is strikingly demonstrated in the opening scenes. This episode is followed by scenes showing the quarrying and loading of silica sand, the principal ingredient of safety glass, and its transportation to the mill, where it is screened, crushed, ground, dried, and conveyed to the glass plant. Operations within the plant incidental to the manufacture of glass are next illustrated, including the mixture of the prepared silica with other ingredients, such as limestone, soda ash, charcoal and water. After being pressed into briquets, the mixture is fed to the furnace; a number of striking scenes show the melting of the mixture inside the furnace, and the sampling, pouring and rolling of the hot glass into a continuous ribbon. Annealing operations, the cutting of the glass into strips and sheets, and the trimming and grinding of the sheets are next shown.

The second reel illustrates the polishing of plate glass, the picking up and turning of the sheets being accomplished by a machine. The cutting of the glass into special shapes is next illustrated. A series of scenes next shows the actual process of making safety glass. The glass sheets are coated with cement, and sheets of cellulose acetate are pressed between them, forming a sandwich which is then pressed by rollers. The sandwich is heated to a temperature of 300 F under 200 lb pressure, then dried, given an acid bath, and inspected. The final steps of grinding and polishing are next shown. The closing scenes show chemists testing the raw materials, and tests of the finished product, including humidity and sunlight tests, and the dropping of steel balls on the glass plates.

Copies of this film in 16-mm and 35-mm size may be had for exhibition by interested organizations by addressing the Bureau of Mines Experiment Station, 4800 Forbes St., Pittsburgh, Pa.

Rubber Springs Described at M. I. T. Conference

Rubber springs that promote efficiency and economy by eliminating vibration were described before the summer seminar on concrete and timber held July 8 at Massachusetts Institute of Technology.

According to F. L. Haushalter, development engineer, B. F. Goodrich Co., rubber springs have met with success in machines ranging up to 500,000 lb, prolonging the life of machinery and easing the strain on the nerves of workers. It was stated that the rubber mountings, called vibro-insulators, may eliminate as much as 90 per cent of the vibration in machinery and extend the longevity of concrete structures in which the machines are housed.

General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning

By F. C. Houghten*, A. B. Newton** (*MEMBERS*), R. W. Qualley***, and Edw. Witkowski†

This paper reports the results of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Committee on Research, conducted under the supervision of the Technical Advisory Committee on Sensations of Comfort and analyzed by the Research Laboratory.

THIS paper deals with the results of a questionnaire sent to 274 employees working in air conditioned offices in Minneapolis during the summer of 1937. These employees were the same as those participating in the daily study reported in the two earlier Laboratory reports.[1, 2]

The questionnaire was given to the employees on September 15 after they had been accustomed to filling out the regular daily questionnaires included in the studies referred to since May 4. Therefore, those participating had more than the usual background in studying and interpreting their reactions to air conditioning, but with the exception of a few of the higher ranking members of the organization who may have been included in the survey, they had no particular knowledge or interest in air conditioning. It is believed that the results of this questionnaire will serve a useful purpose in indicating the reactions of normal persons occupying air conditioned space during the working part of the day, and that they will help clear up a number of questions which are generally conceded to be controversial in the minds of many laymen and some persons more closely associated with air conditioning.

A copy of the questionnaire which was given to employees on September 15 and collected on September 16 is included. No other advice or suggestions were given concerning the returns to be made.

QUESTIONNAIRE ON AIR CONDITIONING

(Answered by 274 Office Employees)

Your cooperation in filling in the daily questionnaire cards, telling us of your reactions to the comfort conditions obtained by the office air conditioning system is very much appreciated. We have obtained a great deal of valuable data from these daily cards, and as we near the end of the cooling season, we also desire your answers to the following questions. We should appreciate as much detail as you can supply:

(1) What difference, if any, have you noticed in your reactions after leaving the office on warm days as compared to the period previous to the installation of air conditioning?

(2) What difference have you noticed between your reactions after leaving the office and reactions of friends or associates who were not in conditioned offices during the day?

(3) General remarks on the effectiveness of air conditioning.

A preliminary analysis of the returns showed certain marked tendency in the replies and included several definite types of answers to each of the three questions. All of the replies are tabulated under the headings of the three questions according to the type of answers.

Immediate Reactions Upon Leaving the Air Conditioned Space

Replies to Question 1, What difference, if any, have you noticed in your reactions after leaving the office on warm days as compared to the period previous to the installation of air conditioning?, were divided into four divisions and sub-divisions:

(1) Felt too warm for short period after leaving.
  (a) Less than 20 min.
  (b) Less than one hour.
  (c) Few hours.
(2) Felt too warm all night.
(3) Felt indifferent to warmth after leaving.
(4) No intelligent answer.

While different persons showed wide variation in the length and character of their statements they could with few exceptions be fitted into these divisions. The percentages given each type of answer to Question 1 are plotted in the chart, Fig. 1. Two hundred seventy-four questionnaires were sent out but only 243 answered this question. The others were not present before the installation of air conditioning; therefore the percentages are based on the 243 answers. They show that 35 per cent generally felt too warm for a period of from less than 20 min to two hours after leaving the cooled space, and that approximately 13 per cent felt too warm throughout the entire night. Another 35 per cent indicated that, in general, they had no noticeable reactions to the hot outside or that they felt indifferent to the outside weather after leaving the conditioned space. Ap-

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[1]Summer Cooling Requirements of 275 Workers in An Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1937, p. 758).

[2]Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. To be presented at the Summer Meeting of the Society, June, 1938.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

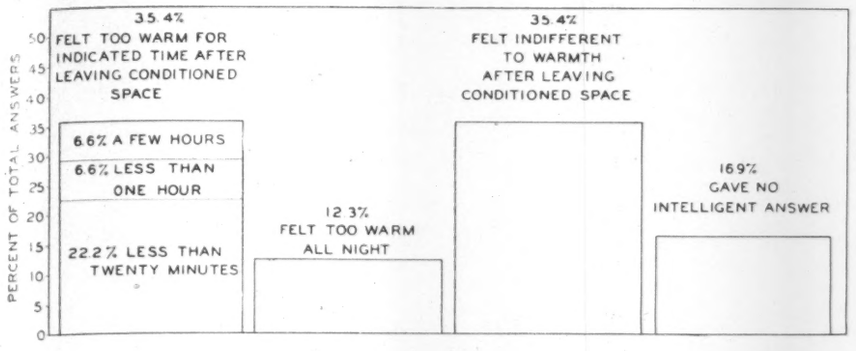

Fig. 1—Answer to Question Number One. What difference, if any, have you noticed in your reactions after leaving the office on warm days as compared to the period previous to installation of air conditioning?

proximately 17 per cent of the replies were not sufficiently to the point to be tabulated in any of the groups.

Many of the answers to Question No. 1 showed very definite characteristics in the feelings of the employees after leaving the air conditioned space. Some of these are of interest in forming a better understanding of air conditioning. A smaller percentage are of interest because they are exceptions. In other cases, the replies were amusing but still educational. A number of typical replies without respect to the percentages of each are quoted:

Was acutely aware of the high temperature immediately upon leaving the building.

A feeling of being quite warm for approximately one-half hour with normal reactions thereafter.

I noticed very little difference. It seemed slightly warmer at first but after being out an hour or so everything was okey.

The sensation of fatigue upon leaving the office has been less noticeable since the installation of air conditioning.

That tired, worn-out feeling on hot summer days is eliminated and I think it enables one to accomplish much more work and still feel comfortable.

I seem to mind the outside heat more. My joints seem to pain me, especially my leg joints,—something like a rheumatism.

Previously the air outside seemed clean and fresh compared to that of the factory. Now, after working in an air conditioned space, the outside air seems stuffy, hot and full of dust and dirt.

On excessively hot days I perspired and noticed the heat more than the rest of the family in the early part of the evening. In fact, I believe it took about four hours for me to become acclimated to the temperature rise.

I noticed quite a change in temperature upon leaving the office on warm days; this change lasted for a short period. During the time previous to the installation of air conditioning my reactions were very much the same upon leaving the office as they were to the air conditions in the office.

The first few months of operation of the system made one feel comfortable inside during the day but in the evening the extreme change of temperature (75 to 100 deg) from inside to outside nearly suffocated one when experienced all at once. But the last few months of raising the inside temperature in the late afternoon (3:00 to 4:30) has made a considerable improvement.

On warm days there is a reaction of being severely warm when one goes from indoors to outdoors, but this is offset by the ability to withstand the heat outdoors better because one is not nearly as fatigued as one would be where no air conditioning is installed. A person can stand the heat better if he is not all tired out from working in the heat all day.

I have not been as nearly fagged out during the hot evenings as I was before the installation of air conditioning.

After leaving an air conditioned office, one is better prepared to withstand higher temperatures outside than if one has been subjected to high temperatures throughout the day.

Before the air conditioning system was installed, working at a drawing board on warm days was very difficult, because perspiration on the arms and hands would spoil a drawing. Now, one does not have this condition to contend with and more work can be accomplished with no feeling of being tired.

I did not feel all worn out by the heat during the day, and although the heat was very noticeable a short time after leaving, I became accustomed to it and was more able to take it as I was not all tired out.

No difference, except that undoubtedly on warm days the outside weather momentarily seems warmer when leaving conditioned space than it would in going outside from unconditioned space.

I seemed to notice the heat more on excessively warm days after leaving a conditioned space than I did while working in an unconditioned space. This sensation of heat lasted most of the evening and night on very warm days.

Having worked in an air-conditioned office all day, the breeze in the evening would appear to be warm or hot. After working in unconditioned offices, and perspiring freely, any evening breeze would appear to be cool.

My skin is so cool that the heat doesn't affect me very greatly.

Heat was very intense and usually lasted until it cooled off after sundown.

The evening heat is less noticeable, because my system has not been under the pressure of heat during the warmer hours of the day.

Reactions as Compared with Those Who Did Not Spend the Day in Air Conditioned Space

Replies to Question 2, What difference have you noticed between your reactions after leaving the office and the reactions of friends or associates who were not in conditioned offices during the day?, were divided into three divisions and sub-divisions:

(1) Stood hot weather better than if had not spent days in cooled space, or than friends who had not spent days in cooled space.

(a) Less complaint of heat.

(b) Less fatigued, less tired, or had more pep.

(2) Did not stand hot weather as well as if had not spent days in cooled space. Friends, who had not spent days in cooled space, stood heat better.

(3) No difference in way they withstood hot nights compared to like feeling when days were not spent in cooled space, or when compared with feelings of friends who did not spend days in cooled space.

Again wide variations in the nature and characteristics of the statements were found in the replies, but with few exceptions the replies could be included in the groupings as given. The percentage showing each type of answer is given in the chart, Fig. 2. Two hundred seventy-four questionnaires were sent out, but only 226 answers to this question were received. Therefore

percentages are based on the 226 answers. A little over 70 per cent indicated in one way or another that they were not as fatigued during hot spells as they would have been had they not worked in air conditioned space during the day time, or as compared with their friends who did not work in air conditioned space. The word *fatigued* naturally only occurred in a relatively small percentage of the total number of replies, but other expressions such as *had more pep, felt less washed-out,* or *less let-down* were included in this grouping. Approximately 13 per cent indicated in one way or another that they did not stand the hot weather as well as they would if they had not spent their time in air conditioned space, or as their friends and associates who did not work in air conditioned space. These answers again were cast in various forms, including statements that they *were more fatigued, more tired, had less pep,* etc. Approximately 15 per cent indicated no marked preference to air conditioning as regards their feelings during the evenings and nights of warm weather. A few characteristic answers which may be of interest are given:

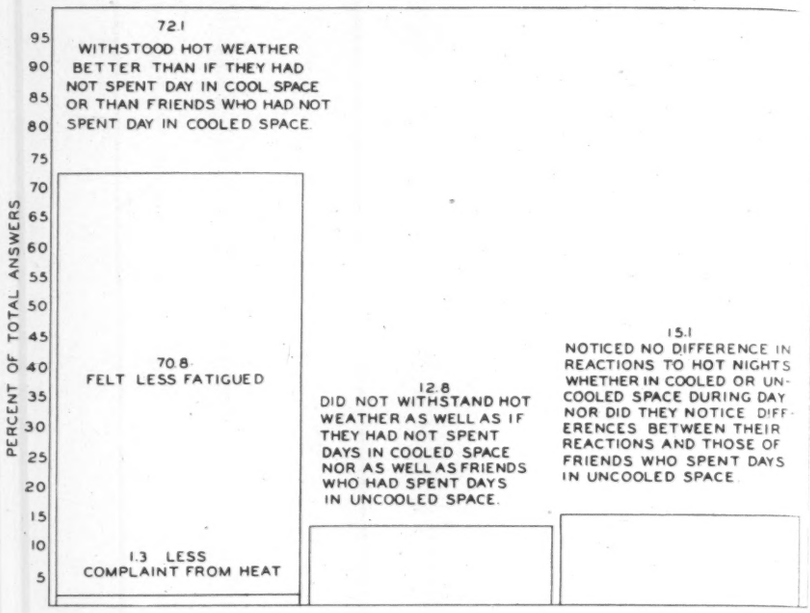
PERCENT OF TOTAL ANSWERS
95
90
85
80
75
70
65
60
55
50
45
40
35
30
25
20
15
10
5
72.1
WITHSTOOD HOT WEATHER BETTER THAN IF THEY HAD NOT SPENT DAY IN COOL SPACE OR THAN FRIENDS WHO HAD NOT SPENT DAY IN COOLED SPACE.
70.8
FELT LESS FATIGUED
1.3 LESS COMPLAINT FROM HEAT
12.8
DID NOT WITHSTAND HOT WEATHER AS WELL AS IF THEY HAD NOT SPENT DAYS IN COOLED SPACE NOR AS WELL AS FRIENDS WHO HAD SPENT DAYS IN UNCOOLED SPACE.
15.1
NOTICED NO DIFFERENCE IN REACTIONS TO HOT NIGHTS WHETHER IN COOLED OR UNCOOLED SPACE DURING DAY NOR DID THEY NOTICE DIFFERENCES BETWEEN THEIR REACTIONS AND THOSE OF FRIENDS WHO SPENT DAYS IN UNCOOLED SPACE.

Fig. 2—Answer to Question Number Two. What difference have you noticed between your reactions after leaving the office and the reactions of friends or associates who were not in conditioned offices during the day?

In general, I was more comfortable, and usually was not quite as tired as a person working in a non-air-conditioned office.

Most people complained about being all tired out and weakened by the heat, but I never had such reactions.

I noticed their apparent consciousness of the heat during the day, whereas, I had not been aware that the day had been uncomfortably warm.

The first hour after leaving the air-conditioned office on a hot day I noticed the heat but soon became accustomed to the warmer air. I noticed friends and associates showed the effects of the hot weather more when they were in an un-conditioned building all day.

They don't mind the outside heat as much as I do. They have as a general rule more pep than I have.

I couldn't talk about the heat of the day. They seem to be more worn out.

I believe I can truthfully say that most of my friends who were not in air conditioned places during the heat of the day were practically worn out by night, while I, being in an air conditioned space all day, was still feeling fine regardless of the perspiring mentioned heretofore.

In general there has not been the complaint this summer as voiced in previous summers. However, the greatest complaint has been poor wives who are domestically confined during the period in which their husbands are air conditioned.

After leaving the office on warm days I was not nearly as tired and I had more ambition; acquaintances who were not in conditioned offices were more tired out at the end of the day and seemed to mind the heat more.

People in non-air-conditioned places were more tired.

I felt the heat more, but since they had to work in it all day, it actually affected me less, in the matter of physical well-being, than it did them. I acquired a slight heat-rash on hot evenings, for about two weeks, but it was probably a condition aggravated by the heat.

I think that my friends who did not work in air conditioned buildings minded the heat more than I did after leaving the office. Air conditioning leaves one in a better frame of mind after leaving work because of being comfortable all day. Persons who do not work in air conditioned places think about the heat all day and by evening they think it's much hotter than it really is.

They were very tired and irritable whereas I felt fine.

I find that I am not as easily irritated as those who do not have the advantage of air conditioning.

After leaving the air conditioned office and returning home, those outside were tired out from the constant heat, while those working in the air conditioned room were still fresh.

At quitting time they were all tired out, not from the work but because of the heat all day. I was not.

In general, reactions were the same except on very hot days. Then friends did not seem to mind the heat as much as I. About 3 or 4 hours in the evening I noticed a difference in reactions.

A definite difference—on hot days followed by hot, close, muggy evenings—I have been much more comfortable and have complained far less of the heat than other members of the family who have not been in conditioned space during the day.

Friends appeared to be every bit as hot and uncomfortable as I was,—in fact, they suffered all day while I was glad to be in the conditioned office.

After a day of comfort in the office, hearing people groan on the street cars going home about the terrific discomfort they had endured at their places of employment, the value of the privilege of spending the day in air conditioned quarters was emphatically indicated. At home, likewise, the complaint of intolerable conditions that made it impossible to accomplish anything during the day was proof of misery we had been spared.

The heat affected them more than it did me. They had made themselves warmer by fretting about the severe heat.

I perhaps noticed the heat outside more than the person who had not been in a conditioned office during the day, but was less tired and could more easily endure the heat for the rest of the day.

Apparently answered previously. I might add that I am much more able to stand the heat of a warm evening than those who have been extremely warm all day.

General Reaction to Air Conditioning

Replies under the heading of Item 3, general remarks on the effectiveness of air conditioning, indicated a logical division into three divisions and sub-divisions:

Analysis of replies to Question No. 3:

(1) General reaction to air conditioning good.
  (a) More comfort and pleasantness.
  (b) Better efficiency.
  (c) Free from colds.
  (d) Better feeling or more pep.

(2) General reactions to air conditioning not good.
  (a) At times, too cool or drafty.
  (b) At times, too warm.
  (c) At times, stuffy.
  (d) At times, too humid.
  (e) Contracted colds or other derangement, including rheumatism or body pains.

(3) Statement not intelligent.

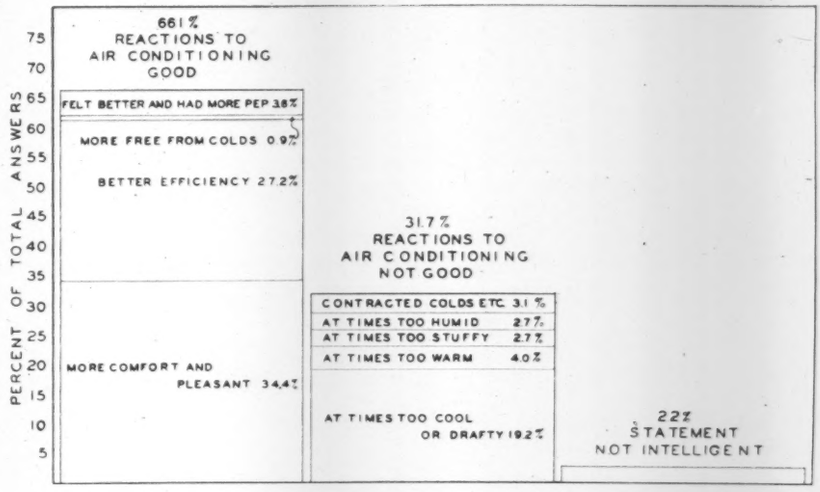

Fig. 3—General remarks on effectiveness of air conditioning

The usual wide variations in the answers were found, but they could be divided into the divisions and sub-divisions given. The chart, Fig. 3, gives the percentages of replies under each classification. Two hundred seventy-four questionnaires were sent out, and 224 statements of reactions to air conditioning were received. Percentages are based on the 224 replies. Approximately 66 per cent showed a satisfactory appreciation for air conditioning. These were sub-divided into approximately one per cent indicating a greater freedom from colds, approximately four per cent having a general feeling of greater satisfaction or more pep, approximately 27 per cent indicating a greater efficiency in their work according to their own interpretation, and approximately 34 per cent spent a generally more comfortable and pleasant summer. Approximately 32 per cent showed a lack of satisfactory appreciation for air conditioning or had a definite criticism of the same. Of these approximately three per cent indicated that conditions were at times too humid, approximately the same percentage that the conditions were at times too stuffy, another three per cent indicated that they were more subject to colds than previously, and four per cent complained that the air conditioned space was generally kept too warm. Of the 32 per cent unsatisfied with air conditioning, there was a rather general complaint covering approximately 19 per cent that the air conditioned space was too cold or drafty. Approximately two per cent answering Question 3 gave answers which were unintelligible.

A few characteristic replies to this question are given:

Appreciate air conditioning only in extremely warm weather; otherwise far too drafty. Further, it does not take care of the smoke in the engineering department which irritates my eyes, nose, and throat. I believe open windows would prove more beneficial.

Was bothered by hay fever out of office. This condition was relieved when in the office. I feel that an air conditioned office has made possible for me a much more comfortable summer.

I think it is very good during warm weather and severe cold weather, but when it is so ideal outside that you want to stay out I prefer to be able to open the windows and enjoy it.

It is a great help to non-smokers who must work between inveterate smokers all day.

I have noticed that the office has been rather drafty at times.

Had to keep my coat on to prevent myself from sneezing.

Air conditioning is okey, except along the south windows,—when the sun shines in here it is hotter than outside. I suggest that you turn on more cold along the windows.

Excellent—keeps personal efficiency at maximum.

My own personal opinion of the effectiveness is that it at least keeps you comfortable one-third of each day and it most certainly is great relief from hot spells in the weather which last weeks at a time.

The effect of air conditioning is beneficial in many ways. It adds greatly to the amount of work that can be produced, especially in the hot summer. It keeps everybody happier and more agreeable. It helps one's health as one is better able to fight off sickness when not all tired out. It offers relief from hay fever and other nose and throat irritations.

The general office temperature is held too low for the comfort of one employed at a desk all day.

After leaving the air conditioned office and entering the warm unconditioned places, perspiration began to flow freely, and for a short time a sort of dizzy feeling would be felt, but would soon wear off.

A person could work all day at the desk without feeling tired or sleepy. No danger of spoiling tracings and papers by excessive perspiration. I think a person working in air conditioned space during the day should try to get into the sun and outdoor heat as often as possible in the evenings, Saturdays and Sundays to induce natural perspiration, as I noticed I felt better the summer of 1937 than 1936 when I tried to avoid the heat after working hours.

In spite of the excessive reaction of heat in the evening, air conditioning is far better than no conditioning, but it could be improved by elimination of draftiness and a certain odor which seems to result from the outdoor air.

I will not soon forget the discomfort of unconditioned space affording only a choice of two evils—perspiration or exasperation

—the latter with a fan blowing one's hair into utter disorder and scattering papers. What ideal air conditions mean to health and comfort and efficiency and enjoyment cannot be condensed into so little space. The passing of the day of drafts from open windows or suffocation with them closed, afternoon fatigue from working in atmospheres lacking proper humidity, air circulation, or at a temperature too high or too low, should mean much to progress and accomplishment for everyone. Summing up, the benefits in health and comfort are immeasurable. With what pride couldn't we show our accomplishment of the optimum air condition to that fellow, Mark Twain, who said, "Everybody talks about the weather, but nobody does anything about it."

Easier to work when office is air conditioned during summer. Would be better if it could be the same in temperature everyday.

More work accomplished during the day than previously and with less fatigue.

More effective work can be done when there is no (or but little) distracting air condition. Due to large effect of many variables—personal health conditions, drinking of water, type of clothing, etc.,—the shock is a delicate, indefinite and negligible factor; too elusive to be pinned down quantitatively. I am, simply, comfortable in conditioned space and uncomfortable in extreme conditions.

Discussion of Results

In general, the answers to the three questions presented in the questionnaire show a considerable percentage of general satisfaction in the experiences resulting from air conditioning. Better than 35 per cent were frank in stating that they felt too warm for a while after leaving the air conditioned space. However, the duration of the hot reaction was in most cases not great, and no great importance was placed on it. An equal percentage showed an entire indifference to or a lack of this hot reaction. Only approximately 13 per cent of the total indicated that their evening and night reactions to the heat may have been increased due to their experience with air conditioning during the day. Undoubtedly, a considerable portion of these lacked satisfactory background on which to base their interpretations. In other words, at least 75 per cent of all inquiries, and a larger percentage of those giving intelligible answers, had no overwhelming or serious reaction upon leaving the conditioned space.

Again, a little over 70 per cent indicated clearly that their unpleasant reactions to the hot nights or unpleasant reactions to a spell of hot weather were lessened by spending the working part of the day in the summer cooled and air conditioned space. When 15 per cent of those who were indifferent to the heat of the night are added to these, less than 13 per cent are found who had a definitely unsatisfactory reaction to the hot evenings and nights, which was charged to air conditioning.

Sixty-six per cent in one way or another indicated definite positive reaction. These reported better feeling, more pep, greater efficiency, greater comfort and pleasure due to spending the working hours or hot portion of the day in air conditioned offices. While it is true that almost 32 per cent stated some objection, these objections were usually limited to certain times, and should not, therefore, be definitely interpreted as a general objection to the satisfaction of air conditioning. They should be accepted rather as objections to occasional lack of perfection which may have actually existed at times in some parts of the extended office space. In this connection, it should be emphasized that on certain days during the study the conditions in small portions of the offices were actually upset so as to make them too warm or too cool for maximum comfort, in order to obtain data for such conditions. It is quite possible that some of these definite conditions which only pertained for short periods of time made more impression on certain individuals than the general satisfaction which they may have felt toward air conditioning in general.

It is felt that the questionnaire serves a useful purpose in indicating the large percentage of general satisfaction to air conditioning and the lack of severity and importance of the hot reaction upon leaving the air conditioned space. It is logical that this should be the case, since it is generally recognized that the unsatisfactory reaction of the human body to an extended hot spell is an accumulative effect. It is well recognized that a person can spend a few hours during the day in a very warm atmosphere without any extended effect. In other words, recovery from short exposure to high temperature is rapid and more or less complete. On the other hand, exposure to hot atmospheres during a hot spell in the summer when one is unable to avoid the heat either night or day is severe and accumulative. The answers to these questionnaires go a long way to satisfy this contention. The questionnaire offers a little data on the contention that a high percentage of cold infections results from exposure to low temperatures maintained in air conditioning. Approximately one per cent indicated a greater freedom from colds, while approximately three per cent indicated a greater incidence to colds. Since it is generally well recognized that summer colds are usually due to infections rather than exposure, it is probable that the few replies received in this connection are of little importance.

The one reply, listed in the sample replies to Question No. 3, charging the feelings of pains in joints and muscles, or rheumatism, to air conditioning is probably another case of the exception which proves the rule. It is probable that in any such extensive group of workers one or two could be found who had recently contracted some form of rheumatism, lumbago, or other source of muscular or joint pains.

Similar studies to those carried on in Minneapolis during 1937 are being planned for other sections of the country during the summer of 1938. The importance of this type of analysis of occupancy in air conditioned space is just being recognized and should be made an important part of future investigations.

Acknowledgment

The Research Technical Advisory Committee on Sensations of Comfort is indebted to the Minneapolis-Honeywell Regulator Co. for their cooperation in providing the facilities of their general office building and the necessary personnel to observe and collect the data for these investigations, and to the Brown Instrument Co. for furnishing the recording instruments used in these studies.

CORRELATING THERMAL RESEARCH

As a part of the efforts of the A.S.H.V.E. Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there is published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, A.S.H.V.E. Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director.*

W. L. Fleisher, *Chairman*
COMMITTEE ON RESEARCH

Heat Transfer in the Condensation of Steam, by Margaret Fishenden. *Engineering*, Vol. CXLV, June 10, 1938, pp. 643-5. Discussion of heat transfer from steam to a metal surface. If a cooling surface is wettable the condensation produces a continuous film of fluid; on non-wettable surfaces the condensation takes place in the form of droplets and the heat transfer may be up to 10 or even 20 times as much as for film condensation. Conditions of the two types of condensation: rough or scratched surfaces, especially when clean and free from grease encourage film condensation; while smooth and dirty surfaces, especially if greasy, encourage drop condensation. But even a rough surface if greasy may give drop condensation while even a smooth surface if freshly cleaned may give film condensation. Both types of condensation may occur either co-existently or alternately, and the changes have sometimes given rise to results which appeared anomalous.

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Heat Dissipation Through an Annular Disk or Fin of Uniform Thickness, by William McGregor Murray, *Journal of Applied Mechanics*, Vol. 5, No. 2, June, 1938, *Trans. A.S.M.E.* Mathematical investigation of heat flow in an annular disk of uniform thickness. The investigation was carried on in connection with the design of fins for increasing the heat transfer in various kinds of heat exchangers and engines. The results might easily be applied to other problems.

•

Temperature Rise in Material of Which the Thermal Properties Vary with Temperature, by J. H. Awberry. *Proceedings Royal Soc. London*, Vol. 48, p. 118, 1936. The partial differential equations governing the flow of heat in a medium for which the thermal conductivity and volumetric specific heat vary parabolically with temperature are reduced to an ordinary differential equation when the medium is semi-infinite in extent.

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Ventilation. Entwurf und Betrieb der Schleuder und Schraubengeblase. Published by Julius Springer, Berlin. Price 13.50 Marks. Instructive and practical book concerned with the manner and the extent to which theory can be reasonably applied to produce sound engineering products in the manufacture of fans. Discussion of centrifugal fans and summaries of performance data for a range of typical fan designs.

•

Removal of Dusts and Fumes; New Departures, by C. W. Price. *Times Trade and Engineering*, Vol. 42, No. 888 (New Series). Factory Welfare Section, p.x. (Feb. 1938). Discussion of new Factory Act, England, with regard to the removal of dust and fumes.

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Dust Control in Industry, by T. C. Angus. *Industrial Welfare*, May 1938. Discussion of hazards to the health of industrial workers caused by the breathing of air-borne fumes and dusts which fall under two headings; first, results of inhaling known poisonous substances; and, second, inhaling other dusts not usually toxic. The new 1937 Factories Act will strengthen existing dust regulations.

•

Appliances for Measuring Radiation from Domestic Heaters, by R. E. Martin. *Engineering*, Vol. CXLV, June 24, 1938, p. 717. Description of radiometer for measuring radiation efficiency of domestic heaters. The heat receiver consists of two sets of six elements interconnected by iron constantan thermocouples in series so that alternate junctions are in opposite faces of the receiver.

•

Stoves for Smoky Fuels, by S. Kasuga. The *Journal of the Fuel Society of Japan*, Section 2, Vol. XVII, Serial No. 188, May 1938, pp. 45-52. Translation in English. This report deals with the firing experiments conducted with eight kinds of smoky coal and one of semi-coke in three types of stoves on the market. Proximate and elementary analyses of fuel used. Tables of carbonization products, fuel gas analyses, ignition time, combustibility and air required for combustion.

•

Research Work on Wind Pressure in Belgium. *Engineering*, Vol. CXLV, May 6, 1938, pp. 498-500, continued *Ibid* June 3, 1938, pp. 638-639. Discussion of development of research work on wind pressure in Belgium. Work on wind action with panels and on the facade of a building is situated at the University of Liege under M. R. Spronck. Wind tunnel experiments and tests on actual structures are centered at University of Brussels with Prof. L. Baes and L. J. Vendeperre.

•

The Estimation of Pipe Delivery from Pitot-Tube Measurements, by A. Fage. *Engineering*, Vol. CXLV, June 3, 1938, pp. 616-7. Mathematical discussion of estimation of pipe delivery from pitot-tube measurements. Also derivation of equation for turbulence correction.

•

Insulation Material, F. M. McClenahan. *Transactions*, Illinois State Acad. of Science, Vol. 29, No. 2, 1936, p. 215. Material for use in insulating buildings and houses is prepared from lightly calcined silicic acid and formed by either extrusion or molding. The product has a heat transmission equivalent to that of prime cork.

•

The Influence of Thickness on the Measured Thermal Conductivity of Fiberboard and Rockwool, by J. D. Babbitt. *Canadian Journal of Research*, Vol. 16, No. 4, April 1938, pp. 82-87. The thermal conductivity of samples of rockwool and fiberboard of various thicknesses (0.5 to 2.0 in.) was measured by means of a hot-plate apparatus. It was found that when surface effects were eliminated the conductivity obeyed Fourier's Law.

•

An Experimental Study of Heat Collapse, by J. S. Weiner. The *Journal of Industrial Hygiene and Toxicology*, Vol. 20, No. 6, June 1938, pp. 398-400. Report of an experimental study of heat collapse. Employees of the Witwatersrand South Africa Gold Mine were subjects. Observations were made on pulse rate, systolic and diastolic blood pressures and the respiratory rate and the oral temperatures while the subjects were working and standing at attention in room temperatures of 94-95 wet-bulb and 96-98 dry-bulb. Some subjects experienced heat collapse.

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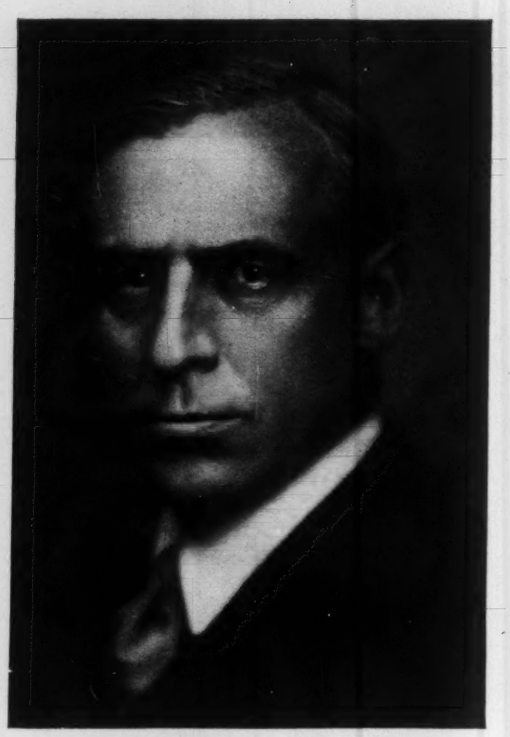
Hospital Air Conditioning, by C. P. Yaglou. The *Journal of the American Medical Association*, Vol. 110, No. 24, June 11, 1938, pp. 2003-2009. Fifth report of the committee established by the *American Medical Association* to study air conditioning. Outline of the objectives and special problems arising in the control of ward atmospheres, with especial reference to operating rooms, post-operative wards, nurseries for premature infants, fever cabinets or rooms, allergen-free rooms and oxygen chambers where air conditioning has proved of value.



Pres. E. Holt Gurney, Toronto, Canada

Proceedings of the Semi-Annual Meeting 1938

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

The Homestead, Hot Springs, Va. June 20-22

AT the Semi-Annual Meeting 1938 held in The Homestead, Hot Springs, Va., June 20-22, over 200 members and guests of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS devoted their attention to a discussion of 13 technical papers during the four sessions. Because of threatening weather two sessions were held on Monday and single sessions on Tuesday and Wednesday.

FIRST SESSION—

Monday, June 20, 9:30 a. m.

Pres. E. Holt Gurney, Toronto, declared the Semi-Annual Meeting 1938 in session and introduced J. A. Donnelly, Largent, W. Va., General Chairman of Arrangements, who welcomed the visiting members and guests to Virginia.

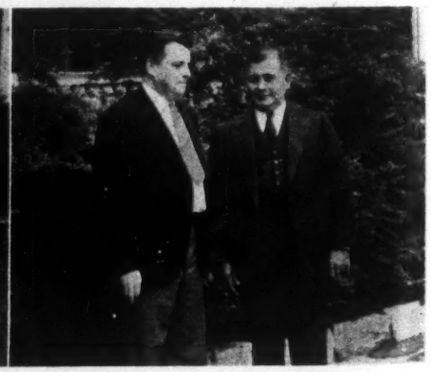
The first paper on the technical program was A Test Method for Air Filters, which was presented by the author, R. S. Dill, Washington, D. C. (Complete paper published in August 1938 A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

Arthur Nutting, Louisville, Ky., pointed out that the test method employed by the Bureau of Standards considers only the initial cleaning efficiency of a filter, whereas the purchaser of a filter desires the highest average cleaning efficiency. He said the Bureau's method gave no indication of the maximum or the average cleaning efficiency.

The next paper on the program was The Flow of Air Through Exhaust Grilles, by A. M. Greene, Jr., and M. H. Dean, which was presented by the co-author, Mr. Dean, Kansas City, Mo., and detailed an investigation to determine coefficients to be used in calculating the air flow through an exhaust grille of modern form in which the work of previous investigators was checked and an improved method demonstrated. (Complete paper to be published in A. S. H. V. E. JOURNAL SEC-

J. F. McIntire, Detroit

Mrs. J. F. McIntire, Detroit, and Mrs. E. Holt Gurney, Toronto



J. R. Vernon and J. J. Aeberly, Chicago

ENJOYING A FEW DAYS IN VIRGINIA

J. A. Donnelly, General Chairman, Committee on Arrangements

TION, *Heating, Piping and Air Conditioning*).

The technical program included two papers relating to air cleaning and the first to be presented was The Condensation Nuclei Content of the Air as Related to Air Freshness, by R. A. Nielsen, East Pittsburgh, Pa., which was given by the author. (Complete paper published in July 1938 A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

Oral comments were offered by Prof. C.-E. A. Winslow, New Haven, Conn., and H. C. Murphy, Louisville, Ky.

In written discussions Lt. Col. W. A. Danielson, Fort Knox, Ky., said that research for the cause of "freshness" in air must be continued and he pointed out that the paper did not consider the Brownian movement of the charge of air which is a tremendous force in nature and will undoubtedly continue as a disturbing element in air research of this character.

Dr. S. L. Warren, Rochester, N. Y., said that it was a common experience to find that, after remaining in a stimulating environment, the stimulation gradually subsided and the "freshness" of the atmosphere disappeared because of the changes in humidity, temperature, oxygen-carbon dioxide ratio, odors, etc., which have a depressing influence.

L. R. Koller, Schenectady, N. Y., said that his findings indicated that the concentration of nuclei is not a usable index of "freshness" of air in conditioned atmospheres, and that the factors of temperature, humidity and content of foreign gases or vapors are probably more important.

Dr. C. A. Mills, Cincinnati, Ohio, expressed the opinion that the problem of determining air "freshness" was essentially a problem for biologic research and suggested that research be sponsored in this fundamental field, rather than on physical methods of conditioning air.

W. L. Fleisher, New York, N. Y., suggested that the paper should include information about the pressures used in the nuclei counter and pressure after expansion, as this might give a clue to the question whether results could have been altered by different methods of counting or applications of pressure.

Lt. Col. W. D. Fleming, Washington, D. C., agreed with the author's conclusion that still air has a high condensation nuclei content and that reduction of this nuclei does not add anything to the "freshness" of air.

The final paper of the first session, Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity Through Filter, by F. B. Rowley and R. C. Jordan, was presented by R. E. Backstrom, St. Paul, Minn. (Complete paper published in August 1938 A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

Oral discussions were given by Messrs. Dill, Fleisher and Lee. In his written discussion, Mr. Murphy said that the development of a dust feeder was an outstanding feature of the cooperative research work and the data developed as to filter efficiency will be most helpful to the Committee which is considering revision of the A. S. H. V. E. Standard Code.

C. V. Haynes

G. W. Barr

Life Membership Certificates Presented

President Gurney announced that weather conditions made it impossible to have the scheduled outdoor sports program and if it was the sense of the meeting that an afternoon session be held, three papers would be presented as follows: Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick; Heat Transfer Through Single and Double Glazing by M. L. Carr, R. A. Miller, Leighton Orr and A. C. Byers; and The Measurement of Heat Transfer by Free Convection from Cylindrical Bodies by the Schlieren Method by L. M. K. Boelter.

It was regularly moved, seconded and voted that an afternoon session be held at 2:00 p.m. and that three technical papers be discussed.

SECOND SESSION—

Monday, June 20, 2:00 p. m.

The first paper reported studies at the University of Illinois and was entitled Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick. The presentation was made by Professor Fahnestock. (Complete paper published in May and June 1938 A. S. H. V. E. JOURNAL SECTIONS, *Heating, Piping and Air Conditioning.*)

M. L. Carr presented the results of investigations made in Heat Transfer Through Single and Double Glazing, by M. L. Carr, R. A. Miller, Leighton Orr and A. C. Byers. (Complete paper published in June 1938 A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

Among the suggestions made by P. F. McDermott, Manville, N. J., for further investigation were the use of wooden spacers and the method of determining air velocities by the Kata thermometer.

Prof. C. M. Humphreys, Pittsburgh, Pa., pointed out the five factors that needed study, which include wind velocities, muntin spacing, shape, and material, and depth of the reveal.

T. S. Rogers, Glen Ridge, N. J., commented on the performance of building materials under test condi-

tions and under installation stresses, and urged the continuation of work.

Prof. E. R. Queer, State College, Pa., discussed the importance of condensation and frost on double glazed windows and the heat transfer characteristics of small and large size panes.

Prof. F. B. Rowley, Minneapolis, Minn., emphasized the care which must be used in applying fundamental constants to practical building construction and gave an analysis of the relationship of test values and calculated coefficients.

R. J. Tenkonohy, St. Louis, Mo., said that a comparison of results of performance of single and double glazed windows under test shows a difference that is most significant in heating work and which becomes even greater in cooling work.

Colonel Danielson emphasized the importance of air motion which was brought out strikingly in the paper and explained the means of preventing condensation with double glazing.

Prof. T. F. Rockwell, Pittsburgh, Pa., presented an analysis of percentage of heat saving based on the average ratio of net wall area to window area and pointed out the cost relationship between the use of double glazing and wall insulation.

W. C. Randall, Detroit, Mich., pointed out the difference in the test results in this report and one published in the 1936 A. S. H. V. E. TRANSACTIONS on Room Surface Temperature of Glass in Windows, and also explained some of the practical difficulties involved in double glazing of steel windows which would be used where inside humidities were high.

The paper on Measurement of Heat Transfer by Free Convection from Cylindrical Bodies by the Schlieren Method, by L. M. K. Boelter, was given by John James, technical secretary of the Society. (Complete paper to be published in A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) In abstracting the paper, it was said that data obtained by means of the Schlieren method are employed in this paper to determine the effect of some simply shaped surroundings upon the convective heat rate from a horizontal circular cylinder. The rate of convective heat transmission may be shown to depend upon the slope of a beam of light as it leaves the thermal field near the surface. The light beam is so directed as to be tangent to the surface at the entrance to the thermal field and the beam slope is determined experimentally. The slope of the light beam as it leaves the cylinder is determined by the deflection from the shadow cast by a cold cylinder on the screen. The deflection of light beam is recorded by the exposure of sensitized paper on a screen.

The effects of four geometrically simple shaped surroundings were observed and arranged to form: (1) a vertical wall parallel to the cylinder; (2) a horizontal ceiling; (3) an intersection of a wall and ceiling parallel to the cylinder; and (4) a square tunnel located concentrically about the cylinder. The results report the ratio of film conductance with the cylinder at any distance from the walls to that with the walls far from the cylinder. These data are shown as a function of the dimensionless ratio as compared with the distance of cylinder from the center of the wall divided by the diameter of the cylinder.

THIRD SESSION—

Tuesday, June 21, 9:30 a. m.

President Gurney opened the meeting and stated that if anyone present desired to reopen the discussion of the three papers presented on Monday afternoon, the opportunity would be afforded to them. As no one expressed a desire to comment on the papers, C. M. F. Peterson, Cambridge, Mass., was introduced by President Gurney and presented the paper, Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson. (Complete paper published in July 1938 A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

At the conclusion of his presentation, some comments were made by Professor Kratz and Paul McDermott.

The second paper in this session, Performance of Surface Coil Dehumidifiers for Comfort Air Conditioning, by G. L. Tuve and L. J. Siegel, was presented by Professor Tuve, Cleveland, Ohio. (Complete paper to be published in A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

Professor Tuve said in making the presentation that this paper is Progress Report No. 4 on the general subject of the heat transfer of surface coils, on which research has been in progress at Case School of Applied Science since 1931. The first two reports dealt with dry coils for air heating and cooling. A year ago at the Swampscott meeting methods for figuring dehumidifying coils by overall coefficients of heat transfer were reported. In the present paper special attention is paid to the use of the individual air-side and refrigerant-side surface coefficients.

Since 1935 at least five important papers have appeared on this subject, most of them in *Refrigerating Engineering.* These papers have probably accomplished two things: (1) They have developed the theory and the formulas for the important relations in coil performance; (2) by a hundred pages of mathematics and tables, they have convinced most engineers that the subject of surface-coil dehumidification is probably not so simple. But complicated things have a way of becoming much simpler with understanding, and the right answers are becoming closer and closer without using complex equations and tables.

It is not difficult to set down the advantages and disadvantages of the various methods for figuring surface coils. This has been done in Table 2. It is desired to be able to rate a coil accurately by a few easy tests, obtain coil performance under any operating condition by a few simple calculations, and to understand the whole process.

Comments were made by C. A. Bulkeley, Philadelphia, Pa., and a series of written discussions was presented as follows: V. T. Kartorie, York, Pa., believed that the authors' work would eliminate temporary expedients and establish a basic system governing the performance of extended surface coils.

B. L. Messinger, Chicago, Ill., presented several questions and a proposed new design for mixing chamber and test house.

R. H. Swart, Beloit, Wis., expressed interest in the tabulation of various methods of figuring coils with the relative advantages and disadvantages and found that the derivation of exit dry-bulb temperatures and surface

temperatures was in agreement with work that he had done.

H. B. Pownall and S. P. Soling, York, Pa., explained that available experimental data for a coil of constant depth and with constant air velocity for varying refrigerant temperature showed that when the refrigerant temperature was lowered, there is a tendency for the leaving dry-bulb temperature to approach the saturation line. They pointed out how the deviation might be small or quite large.

William Goodman, La Crosse, Wis., said that the humidity method described would be subject to errors inherent in the constant surface temperature method plus any error due to the assumption that the curve of leaving air conditions coincides with the constant humidity curve, and questioned the value of the method outlined by the authors.

H. F. Hutzel, Detroit, Mich., believed that the analysis made by the authors does not enable determination of the refrigerant temperature required to obtain a given surface temperature. He said that it was his opinion that a universally applicable equation would have to consider internal tube diameter, velocity of refrigerant, and ratio of extended to internal tube area. He said that the paper offers some valuable ideas which may suggest an accurate solution of a very perplexing problem.

W. L. Knaus, Fort Wayne, Ind., said that the paper presented a simple and useful method of analyzing coil performance and suggested a review of the analysis and tests to establish a range where the humidity method is satisfactory.

Status of Research Work

The Committee on Research held two lengthy sessions and considered the reports submitted by the 24 technical advisory committees. Contracts for cooperative work were renewed with the University of Minnesota to continue investigations on air filters; the University of Illinois Medical School to investigate the effect of atmospheric conditions on cardiac cases; and the University of Pittsburgh to observe the reactions of patients, doctors and attendants to various atmospheric conditions in an operating room.

To assist in launching a three-year investigative program on air distribution, new agreements were approved with the University of Wisconsin, Case School of Applied Science to study the behavior of air discharge from grilles and when funds are available at the University of Illinois.

Coöperative agreements were authorized for work at Princeton University to study problems of sound control and the University of Toronto, Canada, for investigating the structure of insulating materials.

A summary of some work now under way is outlined herewith:

Sensations of Comfort: C. Tasker, *Chairman.*

Negotiations have progressed whereby definite arrangements have been made to conduct comfort studies this summer in the Federal Reserve Building, Washington, D. C., the Public Service Building, San Antonio, Texas, and the Metropolitan Life Insurance Building, New York City.

Treatment of Disease: Dr. T. L. Hazlett, *Chairman.*

Studies are being continued on the effect of atmospheric conditions on cardiac cases at the Medical School of the University of Illinois by Dr. R. W. Keeton. This program is outlined in detail in the Annual Report of the Committee on Research 1937.

A cooperative study has been developed with the University of Pittsburgh Medical School where regular observations are now being taken on the comfort of the patient, doctor and his attendants in the operating room at the Magee Hospital to determine their reactions to various atmospheric conditions. At the same time efforts are being made to assemble the necessary instruments and personnel to determine the anesthetic content of the air in different parts of the room, the bacteria content of the air as well as the desirable physiological facts.

Air Conditioning in Industry: A. E. Stacey, Jr., *Chairman.*

The present objectives of this committee deal with the air conditions prevalent in industrial applications. Steps will be taken immediately to investigate the physiological reactions of individuals to conditions along the effective temperature lines of 85 and 87 deg with men at rest and stripped to the undershirt. These studies will be conducted in the Research Laboratory psychrometric rooms in Pittsburgh. The program is further intended to determine the critical point of heat loss from the human body for lines B, C and E which are represented on Fig. 8, page 74 of THE GUIDE 1938.

Air Cleaning and Atmospheric Impurities: H. C. Murphy, *Chairman.*

The committee is considering a revision to the Society's Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. It is expected that some method of comparing air filter efficiencies by the use of a light meter will be considered in these revisions.

It is contemplated that one phase of the program now being outlined by the committee personnel will include a study in establishing the reduction in bacterial content of air before and after cleaning with different methods of application Further it is felt that some study of the characteristics of the bacteria involved is essential particularly in view of the more recently accepted demonstration of the fact that pathogenic bacteria may be air borne.

A. W. Williams
Columbus, O.

E. J. Febrey
Washington, D. C.

W. W. Cox
Seattle, Wash.

J. J. Nesbitt
Philadelphia, Pa.

J. J. Finan
Chicago, Ill.
ELECTED TO LIFE MEMBERSHIP

Radiation and Comfort Winter and Summer and Effect of Varying Humidity on Radiant Heating and Cooling: J. C. Fitts, *Chairman.*

As a result of a series of preliminary conferences and by the exchange of correspondence with the committee personnel a tentative scope of the work to be projected is as follows: (1) A study based on primary sense reactions of the relative comfort derived from rooms with normal window arrangements from convected heat and from heat supplied by both convection and direct radiation; (2) A study of the effect of radiation in governing the sense of warmth of a person in an occupied space, together with the effect of dry-bulb temperature, moisture content and movement of the air already studied by the Society and included in the Effective Temperature Index or the Comfort Chart; (3) A study of the physiological reactions of persons, including skin temperature, comfort and health measurements, to rooms heated with and without radiation; and (4) A study of the relative effect on comfort and physiological reactions by radiation of varying wave-length, and the effect of the relative humidity of the air within the room on the effectiveness of such radiation.

As an immediate objective the committee is considering the initiation of studies covering item 1 to be accomplished at the Research Laboratory.

Weather Design Conditions: T. H. Urdahl, *Chairman.*

The program as outlined by the Chairman of this committee has suggested the following tentative plan for action as a result of a series of conferences and exchange of correspondence between the committee personnel: (1) That a bibliography be prepared and circulated to the committee members, outlining the work already done and the manner in which the various data were collected; (2) That the factors which were not considered in any of these studies be brought out as problems for future consideration; and (3) That some method be adopted whereby a recommended procedure of collecting weather data be established.

The chairman of this committee has reported that this program is now ready for execution and it is expected that the necessary data will be assembled and correlated at the Research Laboratory.

Radiation with Gravity Air Circulation: M. K. Fahnestock, *Chairman.*

The research work of this committee conducted at the University of Illinois may be divided into two parts: (1) that which has or may be done in a warm wall test booth; and, (2) that which may be done in a room heating testing plant, where test conditions simulate actual service conditions.

Under part 1 an extensive study of the inlet and outlets of convector heater cabinets with reference to their effect upon the heat output or capacity of the units has just been completed. These data will be presented in a technical paper. Future work in the booth will probably include tests on lightweight direct cast-iron radiators.

Under part 2, the remodelling whereby the test plant was enlarged to provide a test rom 15 ft x 18 ft x 8 ft 6 in. in place of the two former 9 ft x 11 ft x 9 ft test rooms is practically completed. The first tests will be to correlate the performance of conventional tubular steam radiators and convectors in the 15 ft x 18 ft test room with their previous performance in the 9 ft x 11 ft test rooms. Future work will include studies with conventional direct cast-iron radiators, lightweight direct cast-iron radiators, and convectors, both free standing in the room, and recessed under windows in exposed walls.

Heat Transfer of Finned Tubes with Forced Air Circulation: G. L. Tuve, *Chairman.*

Work is being resumed in checking the various methods for calculating dehumidifying coils. Special attention is being given to the determination of surface heat transfer coefficients which may in turn be combined to give the overall coefficient for any desired operating condition.

The entire program is directed toward the formulation of uniform practices, which can then be recommended as the basis for a standard Test Code for rating and calculating surface coils. Five subdivisions of the general problem are covered in the present plans for experimental studies and correlations: (1) Variation of air-side surface coefficients of both dry and wet coils; (2) Effect of water velocities on the water-side coefficient of both refrigerating and heating coils, connected in counterflow and in cross-flow; (3) Improvement of methods for measuring refrigerant-side surface coefficients of direct-expansion coils; (4) Effect of coil depth or number of rows of tubes, and performance of very deep coils, (6 to 10 rows); and (5) Relation of heat transfer to air friction and air velocity.

Cooling Load in Summer Air Conditioning: J. H. Walker, *Chairman.*

A preliminary report covering the research investigations observed at the Research Laboratory has been distributed to the committee personnel giving fragmentary data on a few walls in which the time lag due to solar radiation has been noted. A complete report of the program which is contemplated for study by the Research Laboratory this summer has been submitted to interested individuals and if financial assistance is obtained it is contemplated that the work can start immediately.

Solid Fuels: W. A. Danielson, *Chairman.*

The general problem of this committee deals with utilization of fuels with the ultimate objective to provide better service to the consumer. The scope of work thus far outlined seems to emphasize the importance of collecting facts from recognized authorities and further attempting to correlate the work of other agencies doing investigative work in this field.

One of the important problems to be considered indicates that in the solid fuel field there is a seeming lack of approved test methods and means whereby comparative standards in performance of equipment can be evaluated. The importance of mutual education from the consumer's viewpoint will be emphasized with regard to cleanliness, convenience and comfort studies. Utilization of solid fuels has been generally classified by the committee as dealing with (1) equipment, (2) method of firing, (3) drafts, and (4) fuel and ash handling.

Summer Air Conditioning for Residences: M. K. Fahnestock, *Chairman.*

The research work of this committee conducted in the Research Residence has been in cooperation with the University of Illinois and the National Warm Air Heating and Air Conditioning Association. This study is to be continued during the summer of 1938 with the following objectives: (1) to obtain more information on the practicability of using a small capacity condensing unit in severe warm weather; (2) to obtain more information on the feasibility of cooling the first floor during the day and the second floor at night; (3) to obtain data in regard to the rate at which the temperature of the air in a space, where cooling has not been in progress, can be reduced with several different cooling capacities; and (4) to study a possible method of alleviating the unsatisfactory fluctuations in indoor relative humidity which occur in mild weather in a residence, where the dry-bulb temperature is maintained by starting and stopping the condensing unit with a dry-bulb temperature thermostat.

Air Distribution and Air Friction: J. H. Van Alsburg, *Chairman.*

A comprehensive program for investigations in this field has been outlined with the work to be accomplished at various cooperative institutions and the Research Laboratory.

At the University of Illinois it has been suggested that tests be conducted in their 15 x 18 ft cold room where the walls, ceilings or floors may be heated or cooled in order to determine the advantage of location of air outlet with specific regard to supply and return.

The program contemplated at the Case School of Applied

Science involves a comparison of instruments used for air velocity measurement and the behavior of air discharge from grilles into large unconfined spaces.

At the University of Wisconsin a cooperative project is being considered which will consist of testing the effect of approaches to an air outlet and possibly further tests in large rooms with variable shaped openings.

Heat Requirements of Buildings: P. D. Close, *Chairman.*

As the GUIDE is materially lacking in data pertaining to heat losses into the ground, it has been recommended that a study be conducted at the Research Laboratory to establish proper heat loss coefficients. Tests have been planned to determine the heat loss into the ground by use of a suitable heat flow meter. The heat loss through top floor ceilings and under pitched roofs has also been considered by this committee and it is expected that a revised method of calculating these losses will be submitted to the Guide Publication Committee.

Air Conditioning Requirements of Glass: M. L. Carr, *Chairman.*

The work of this committee is actively progressing as a result of the program established last year with substantial earmarked funds to conduct current investigations. A technical paper reporting the work thus far accomplished by this group was presented at the Semi-Annual Meeting of the Society entitled Heat Transfer Through Single and Double Glazing.

Insulation: W. A. Danielson, *Chairman.*

A comprehensive program of research activity was developed by this committee a year ago with particular interest displayed in the development of codes for testing conductivity and heat transfer. Some of the subjects dealing with the fundamental aspects of insulation constitute the general work of the committee and may be enumerated as as follows: (1) the effect of regain moisture on the conductivity of insulating materials, (2) the effect of condensed moisture on the conductivity of insulating materials in a structure, (3) effect of the thickness of the insulating material accepted for test and the conductivity value obtained, (4) effect of condensation on the surface of reflective insulation, (5) effect of fiber size and fiber arrangement within an insulating board on its conductivity, (6) relation between the heat capacity of an insulating material and the rate of heat penetration through a structure in which it is installed, and (7) the degree of settling found for different fill materials after they are placed in a structure.

Sound Control: J. S. Parkinson, *Chairman.*

It is the consensus of this committee that a code is both necessary and desirable for testing and rating the noise of air conditioning equipment. A suggested code has been outlined and submitted to the members of this committee for detailed consideration. A request has also been forwarded to various air conditioning manufacturers soliciting their cooperation in furnishing the committee with any available data now being used in predicting the noise levels in a room and applying this information to the selection of equipment and application of sound absorptive materials.

Cooling Towers, Evaporative Condensers and Spray Ponds: B. M. Woods, *Chairman.*

Primary factors affecting this type of equipment have been under consideration by this committee. In connection with cooling tower performance the problems for study have been generally classified as those dealing with the gross factors usually considered for a given tower, such as wind velocity, wet-bulb temperature of air striking the tower, temperature of water on the tower, and loading of water on a tower in gallons per minute per square foot of area.

The cooling effect of various thicknesses of water maintained on roof decks is being studied by this Committee and during the current summer it is expected that various observations will be made on test cubicles to ascertain actual surface temperatures.

Corrosion in Steam Systems: A. R. Mumford, *Chairman.*

This committee is actively engaged in developing a program to determine the probable effect of the rate of condensation upon the oxygen content of a condensate leaving a radiator. This work is a continuation of that outlined in the Annual Report of the Committee on Research 1937.

Corrosion in Air Conditioning Equipment: A. E. Stacey, Jr., *Chairman.*

At the annual meeting of the Society in New York last January a memorandum was reviewed covering the work thus far reported by this committee. Since that time additional correspondence has been exchanged between the committee members and it is expected that a report will soon be released dealing with this subject.

FOURTH SESSION—

Wednesday, June 22, 9:30 a. m.

On Wednesday morning the fourth session was called to order by President Gurney and a report of the tellers on the ballots for the A. S. H. V. E. Standard Code for Testing Stoker-Fired Steam-Heating Boilers showed 348 legal ballots, of which 345 voted approval of the Code and three disapproved. A series of four papers on summer cooling studies was announced.

The first speaker, C. Tasker, Toronto, Canada, gave Cooling Requirements for Summer Comfort Air Conditioning in Toronto. (Complete paper published in June 1938 A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

Observations made at the Agricultural and Mechanical College of Texas were given in the paper, Seasonal Variations in Effective Temperature Requirements, by F. E. Giesecke and W. H. Badgett, which was presented by Mr. Badgett. (Complete paper to be published in A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

Additional data collected at Minneapolis, Minn., were further analyzed in two papers: Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson, which was presented by Mr. Houghten. (Complete paper published in July 1938 A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*); and in General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning, by F. C. Houghten, A. B. Newton, R. W. Qualley and Edward Witkowski, which was presented by Mr. Newton. (Complete paper published in August 1938 A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Contioning.*)

Discussions were given by L. T. Avery, Cleveland, Ohio, M. S. Cooley, Washington, D. C., W. R. Rhoton, Cleveland, Ohio, W. L. Fleisher, W. E. Zieber, York, Pa., F. R. Bichowsky, Ann Arbor, Mich., and F. C. Houghten.

Amendments to Research Regulations

The amended Regulations Governing the Committee on Research were presented and on motion of Mr. Fleisher, seconded by Mr. Tasker, it was unanimously voted to approve the following articles:

Article II, Organization—Section 2. Chairman and Vice Chairman, Committee on Research—A Chairman and a Vice

Chairman of the Committee on Research shall be appointed each year by the President of the Society immediately after election, such appointments to be made from the membership of the Committee on Research. The member appointed Chairman shall have served on the Committee not less than one (1) year.

OMIT SECTION 3. Section 3. Vice-Chairman—A Vice Chairman shall also be elected in the same manner and for the same period as stated for the Chairman in Section 2.

Section 4. Research Executive Committee—There shall be a Research Executive Committee consisting of the Chairman and the Vice Chairman of the Committee on Research and three (3) other members of the Committee on Research, appointed by the Chairman immediately following the Annual Meeting.

Article III, Duties of Officers—Section 1. Chairman—The Chairman of the Committee on Research shall preside at all meetings of the Committee and the Research Executive Committee. He shall be an ex-officio member of the Council. He shall approve all bills against Research activities before payment is made.

Section 3. Director of Research Laboratory—The Director of the Research Laboratory shall be in direct charge of the Laboratory under the direction of the Chairman of the Committee on Research. He shall be responsible to the Committee on Research, and shall cooperate with the Chairman of the Committee in every way possible. He shall approve all bills against Research activities before payment is made.

Article IV, Duties of Committees—Section 1. Committee on Research—The Committee on Research shall have general charge of all Research activities including the making of the necessary contracts for rental or purchase of equipment or materials. This Committee shall select and engage a Director of the Research Laboratory, subject to the approval of the Council and such assistants of the Research Laboratory and the Committee on Research as may be required, or the authority for the selection and employment of the Laboratory assistants may be delegated to the Director of the Laboratory by authority of the Committee if the Committee so desires. The Committee shall determine all salaries, approve all cooperative agreements and determine the order in which the subjects shall be investigated by the Research Laboratory. Any proposed expenditure of Research funds outside of the approved budget shall be approved by the Committee on Research before the expenditure is made.

Article V, Government—Section 3(a). Approval—All acts of the Committee on Research other than those specifically authorized shall be subject to review by the Council.

Section 7(a). Payment of Bills—All bills against Research activities shall be presented to the Director of the Laboratory and to the Chairman of the Committee on Research, who will present approved bills to the Secretary of the Society.

The Secretary shall check and, if correct, approve for payment all items when covered by the budget, or emergency expenditures approved in writing by the Chairman of the Committee on Research.

A summary of these accounts shall be made at least every thirty (30) days on a form approved by the Council and presented to the Treasurer of the Society, who shall draw a check against the Research Fund, payable to the American Society of Heating and Ventilating Engineers, for the total amount of the approved bills, which check shall be deposited in the Secretary's special account. The Secretary shall draw against this amount in settlement of all approved expenditures.

Section 7(b). The Secretary shall have the authority to pay salaries, traveling expenses, and petty cash in accordance with the budget.

Section 8. Accounts—The book of accounts shall be kept at the headquarters of the Society, and shall be audited each year at the time the Society books are audited and by the same certified public accountant who examines the Society accounts. The accountant's report shall be presented at the Annual Meeting of the Society by the Chairman of the Committee on Research as the financial report on research activities for the year.

Amendments to Constitution

The Secretary reported the receipt of a petition signed by ten members for the amendment of Article C-V, Section 1, of the Society's Constitution. President Gurney asked that the original article and the amendment be read by the Secretary.

Article C-V—The Council:

Section 1. The affairs of the Society shall be managed by a Board of Directors chosen from among the persons entitled to vote, which shall be styled "The Council." It shall consist of the President, First Vice-President, Second Vice-President, Treasurer, the last Past President and twelve (12) elected members, four (4) of whom shall be elected each year to hold office for three (3) years. The Secretary may take part in the deliberation of the Council, but shall have no vote therein.

To be amended as follows:

Section 1. The affairs of the Society shall be managed by a Board of Directors chosen from among the persons entitled to vote, which shall be styled "The Council." It shall consist of the President, First Vice-President, Second Vice-President, Treasurer, the last Past President and twelve (12) elected members, four (4) of whom shall be elected each year to hold office for three (3) years. The Chairman of the Committee on Research shall be an ex-officio member of the Council. The Secretary may take part in the deliberation of the Council, but shall have no vote therein.

Resolutions

J. F. S. Collins, Jr., Pittsburgh, Pa., offered the following resolutions of appreciation:

Before closing this pleasant instructive meeting, we the members of the American Society of Heating and Ventilating Engineers assembled in Hot Springs, Va. for the Semi-Annual Meeting 1938, wish to thank those who are responsible for making it an outstanding success, therefore,

Be It Resolved, That we are indebted to the Committee on Arrangements who under the able leadership of their general chairman, J. A. Donnelly, and his committee chairmen, have worked so faithfully and efficiently.

That, we appreciate the messages brought to us by the very able speakers at our technical session and at our banquet to stimulate us through the coming months.

That, we enjoyed the scenic and inspection trips provided by the Arrangements Committee.

That, we are appreciative of the service and courtesies extended by the management of The Homestead.

That, we thank the press for its generous support in giving space and publicity to the activities and to the papers presented at our sessions.

Be It Further Resolved, That to all those who have contributed toward making this meeting a success, we wish to express our sincere appreciation.

On motion of Mr. Collins, seconded by Mr. Fleisher, it was unanimously voted to adopt these resolutions.

President Gurney asked if there was any unfinished business to be brought before the meeting or any new business and, in the absence of any further business, a resolution to adjourn was adopted.

Entertainment and Sports Program

A card party and tea was held for the ladies in the Palm Room of The Homestead at 2:30 on Monday afternoon and eight tables of players participated. Those who were rewarded for their skill received loose-powder compacts: Mrs. C. M. Humphreys, Pittsburgh, Pa., Mrs. R. L. Blanding, Providence, R. I., Mrs. J. F. Collins, Jr., Pittsburgh, Mrs. M. W. Bishop, Chicago, Ill., Miss B. B. Mahon, Scranton, Pa., Mrs. G. F. Miller, Washington, D. C., Mrs. C. E. Price, Chicago, and Mrs. Arthur Nutting, Louisville, Ky.

At 3:00 p.m. the children assembled at The Homestead pool for a swim and the aquatic program was directed by R. McC. Johnston. Those who took part were Jean, Arthur, Jr., and David Nass, Pittsburgh, Pa., Stephanie Doster, Litchfield, Conn., Ruth and Marjorie Nelson, Madison, Wis., Charles Farnham, Buffalo, N. Y., and Howard Meinke, Baldwin, L. I., N. Y.

It was observed that ping pong had its devotees.

In the evening at 7:30 p.m. a Get-Together Dinner was held in the Empire Room and a real old-fashioned southern chicken dinner was served. Hill Billy music was enjoyed during the dinner hour and then dancing continued in the Crystal Room. President Gurney made special awards to Mrs. F. Paul Anderson and Mrs. Virginia F. Anderson, Lexington, Ky., M. W. Bishop, Chicago, Ill., and G. H. Tuttle, Detroit, Mich.

Tuesday morning the ladies participated in a golf putting contest and the winners were Mrs. H. C. Murphy, Louisville, Ky., Mrs. M. F. Blankin, Philadelphia, Pa., Mrs. E. W. McMullen, Joplin, Mo., and Mrs. D. J. Van Marle, Buffalo, N. Y.

At 2:00 p.m. the golfers teed off on The Homestead course and competed for the Research Cup and the Eichberg Memorial Trophy.

At the conclusion of the tournament Tom Brown, Chicago, had possession of the Research Cup, and the Illinois Chapter team, consisting of Tom Brown, E. J. Gossett and E. W. Rietz, took the Eichberg Cup back to Chicago for the second successive year. The winner of the flag tournament was E. J. Gossett, Chicago. The low gross score was made by Arthur Nutting, Louisville, Ky., and other awards were made to G. F. Miller, Washington, D. C., most par holes; M. F. Blankin, Philadelphia, least number of putts; E. W. Rietz, Chicago, most one putt greens; W. J. Osborn, New York, most consecutive pars; W. L. Fleisher, New York, par on 13th; M. W. Bishop, Chicago, most 3's; E. W. McMullen, Joplin, Mo., most 4's; H. D. Jackes, Syracuse, N. Y., most 5's; M. W. McRae, Chicago, blind bogey; C. E. Price, Chicago, and D. S. Boyden, Boston, selected holes; and A. E. Wake, Chicago, guest prize.

At 2:00 p.m. a group of 27 members left the hotel in cars to inspect the mill of the West Virginia Pulp and Paper Co. at Covington. Under the direction of Mr. MacDonald guides escorted members through the various departments of the plant. At 2:30 p.m. horses and carriages drove to the door and 43 members and ladies enjoyed a two-hour carriage ride through the winding roads to Dunns Gap.

At 7:30 p.m. the Semi-Annual Banquet was held in the Empire Room with over 200 present. Those seated at the head table were Prof. E. O. Eastwood, Mr. and Mrs. E. Holt Gurney, Dr. C. M. Newman, Mr. and Mrs. J. F. McIntire, Messrs. A. J. Offner, D. S. Boyden, W. H. Driscoll, J. A. Donnelly, C. V. Haynes, and G. W. Barr. Professor Eastwood of Seattle, a native Virginian, acted as toastmaster. President Gurney presented life membership certificates to C. V. Haynes, Philadelphia, and G. W. Barr, Villa Nova, Pa. W. H. Driscoll, Syracuse, past president of the Society, presented the president's memory book to D. S. Boyden, Boston, who responded briefly and expressed his appreciation to the Council and his happiness in having served the Society as president.

Professor Eastwood introduced Dr. C. M. Newman, of the Department of English, Virginia Polytechnic Institute, who gave a most entertaining and humorous talk on the subject of Some Elements of American Humor. W. L. Fleisher presented the Research Cup to Tom Brown and the Eichberg Cup to Elmer Rietz. After the dinner and the short speaking program dancing was enjoyed in the Crystal Room.

On Wednesday several informal golf matches were played on the Homestead and Cascade Courses under almost ideal weather conditions and the tennis enthusiasts had their innings on the fine Homestead courts.

The following Committee on Arrangements was in charge of activities: J. A. Donnelly, *General Chairman;* Perry West, *Reception;* I. B. Helburn, *Registration;* E. W. Klein, *Finance;* J. K. Peebles, Jr., *Publicity;* J. A. Shanklin, *Banquet;* L. F. Nordine, *Golf;* R. McC. Johnston, *Entertainment and Sports;* and A. Buenger, *Ladies.*

It was a meeting that will be remembered with pleasure by all who attended.

Atlanta Chapter Participates in Engineers Outing

June 10, 1938. The Atlanta Chapter of the Society met with the local associations of the *American Institute of Electrical Engineers* and the *American Society of Mechanical Engineers* at the East Lake Country Club to enjoy the annual Engineers Outing.

The program included golf in the afternoon, followed by a dinner dance, which was attended by the wives and guests of members. Specifications for a Miss A. S. M. E., Miss A. I. E. E. and Miss A. S. H. V. E. were read and Mrs. L. L. Barnes was chosen Miss A. S. H. V. E.

Secy. H. K. McCain has reported that the newly elected officers of each Society were announced, with T. T. Tucker reading the list for the A. S. H. V. E. Atlanta Chapter.

H. J. Rose Joins Westchester Distributor

Announcement has been made of the association of Howard J. Rose with Suburban Air Conditioning Corp., White Plains, N. Y., distributors of Carrier Corp. in Westchester County, New York, and eastern Fairfield County in Connecticut. Mr. Rose will manage sales and development activities in a program planned to broaden service to jobbers, contractors, and dealers in the heating and refrigerating field.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.